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Strength Characteristics of Plastic-Bonded Plywood

By G. B. PARSONS,[1] NEW YORK, N. Y.

New methods of molding plywood by the bag process, combined with new adhesives, have opened a wide range of possibilities for the use of plywood in aircraft construction. This paper explains some of the more important advantages of plastic-bonded plywood and presents some of the pertinent basic-strength characteristics. The prediction is made that in the near future with gluing technique advancing rapidly, a thin metal covering glued to a plywood core will be used to great advantage in airplanes having high wing loadings. Such construction will combine the high axial strengths of the metals with the good buckling characteristics of the thicker plywood construction.

WHEN a few years ago engineers made possible the production of aluminum in quantity at low cost the aircraft designer decided that here finally was the perfect material for which he had been searching. So great was the swing from wood to aluminum that the former was almost completely forgotten.

With the present situation demanding immediate production of an almost unbelievable quantity of planes, the aviation industry finds itself confronted with a problem of increasing scarcity of material. All around one hears the cry, "Give us the aluminum and our production will go up 50 per cent." This request is an impossibility so aircraft designers have had to turn once again and rationally investigate the part that wooden structures can play in the present crisis. The trend in most cases has been toward the use of plastic-bonded plywood for nonstructural parts; such as flaps, ailerons, tail cones, or any compound-curvature parts that have presented trouble to the manufacturer.

New methods of molding plywood with new adhesives have opened an entirely new field in which possibilities are unlimited and in which there are very few available data on basic-strength characteristics.

The purpose of this paper is to discuss some of the many advantages of this plastic-bonded plywood and to compile some of the pertinent basic-strength characteristics.

The strength-to-weight ratio of wood has long been a talking point of engineers interested in wooden construction. Wood by itself had numerous disadvantages which limited its application and helped cause the trend toward complete aluminum structures. Wood alone has a low resistance to splitting along the grain, a low tensile and compressive strength across grain, a rather low shearing strength, and an affinity for moisture. These disadvantages had to be overcome to a large degree before wood could again compete with the low-density metal alloys. Plywood appeared to be the answer to most of the problems, but plywood by itself could not be forced into compound curvatures or very sharp single curvatures in one direction without setting up initial internal stresses which made it unreliable. This led to the development of the so-called bag-molding process as used today.

Bag-Molding Process

In this process, thin veneers are placed either in an inside or outside mold having the contour desired and a rubber blanket placed over the entire lay-up and clamped around the edges. The mold is then placed in a tank and heat and pressure applied. Usually this heat and pressure are obtained by steam which forces the blanket down upon the veneers and distributes the pressure evenly over the entire surface. The temperature and time required in this tank are dependent upon the cycle of the glue used and the thickness of the panel. By use of these thin veneers, the internal stresses introduced are negligible. The part upon removal from the tank then usually requires only a simple finishing operation and the assembly of attaching fittings.

The glue used in this process is either in the form of thin sheets or a liquid glue spread on the veneers. The use of Tego film which is of the sheet type, has been found to be very satisfactory since the amount of glue may be more easily controlled. This controlled glue line is quite important if panels are to be spliced together for if there is too deep a penetration of the glue into these thin veneers, a problem of getting a bond on scarfed joints arises.

The glues used are of one of two main categories: either thermosetting or thermoplastic. Thermosetting glue is glue which experiences a chemical change under heat in which water is given off. It may be compared for analogy to cement, in that once it is set by the condensation of this water it can never be made liquid again, even under boiling conditions.

Thermoplastic glues, on the other hand, experience no chemical change but rather jell upon setting and can be transformed again to their original state by heat. They might well be likened to butter or wax.

Balanced Construction for Plywood

In the construction of plywood by this process, it was soon found that a balanced construction was necessary, because of the different characteristics across and along grain directions. By a balanced construction is meant that veneers be symmetrically located in respect to thickness, grain direction, and species about the center line of the panel. If this procedure is not followed, warping of the panel will be evident immediately upon removal from the tank.

By the use of thin veneers and careful lay-up, a panel that is practically isotropic may be developed. This lay-up causes a decrease in allowable tensile and compressive values from the plain wood when forces are parallel to the grain but increases greatly the shear allowables of the panel. In practice, since there is usually a predominating stress occurring at a certain section, the veneers can be arranged in the most desirable manner to accommodate this stress.

Taking for illustration the design of a monocoque wing, it can be demonstrated how easily adaptable a plywood construction is. By careful design of veneer construction, it is possible to have highly predominating grain direction on the top and bottom skins where the tension and compression stresses due to bending are the major consideration. Then as we move along the contour and more nearly approach the neutral axis, longitudinal- or spanwise-grained veneers can be tapered out until a lay-up of approximately as many spanwise- as chordwise-grained veneers exists.

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At this point the tension and compression stresses are at a minimum, and the shear stress is the design criterion. Thus by careful designing this anisotropic material can be made to work very efficiently.

Moisture Content of Woods Used in Plywood Construction

An important correction that must be made in all computations for allowable strengths of wood members is that of moisture content. As a piece of wood becomes drier, the majority of its strength characteristics are increased rapidly. The obvious conclusion is to use very dry woods and to put a protective coating on the surface in order to keep this moisture content as low as possible. However, experience has shown that the stress at the proportional limit increases at a greater rate than does the modulus of rupture or the modulus of elasticity. The material seems to lose ductility in a dried-out condition, and there is a very limited plastic-flow range before failure takes place. It is evident that local concentrations of stresses in a dry material offer a much more difficult problem.

From the standpoint of gluing, if the moisture content is too low a starved glue joint is liable to occur. It was determined (1)[2] that a moisture content of approximately 7 to 9 per cent was the most efficient in affording a good bond between the veneers. If the moisture content of the wood is quite high a longer time is necessary in the tank operation. This may result in a boiling out of moisture from the cellular structure of the wood with a consequence of low ductility and a brittle material resulting. This is not a question of tensile strength since it has been shown, in tests made on European woods by Gerngross at the Technical University of Berlin (1), that the effect of heating to 284 F for varying periods of time had little effect on the tensile strength.

There have been numerous articles written on determination of strength characteristics for different moisture contents. In a bulletin published by the U. S. Department of Agriculture (2) an equation method is developed which affords a fairly accurate means of estimating strength characteristics for a desired moisture content from values obtained for two other moisture contents. The formula as given is

$$\log S_D = \log S_C + (C - D) \log \frac{(S_B + S_A)}{A - B} \ldots\ldots [1]$$

where A, B, C, and D are percentages of moisture content; S_A, S_B, S_C, S_D are corresponding strength values, S_A and S_B being known strength values for moisture content A and B from tests. Term S_C equals either S_A or S_B, and C equals the corresponding value of A or B.

Using this relationship and values of A, B, S_A, and S_B from Table 1 of the same reference, the author has here plotted the strength - moisture content curves for five woods commonly used in the aircraft industry, i.e., birch, spruce, red gum, basswood, and yellow poplar. These resulting curves for modulus of elasticity, modulus of rupture, and compression parallel to the grain are given in Figs. 1 to 5, inclusive. The values for African mahogany, as plotted in Fig. 6, were obtained from a paper by G. E. Heck (3). These graphs are plotted with 12 per cent moisture content as unity, since this is more or less the standard adopted. To correct to 12 per cent moisture any specimen with known moisture content it is necessary only to divide the value obtained at the known moisture content by the value read from the curve corresponding to the strength characteristic in question.

Modulus of Elasticity

One of the first things an engineer must know about a substance is its modulus of elasticity. In this regard, wood presents a little

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

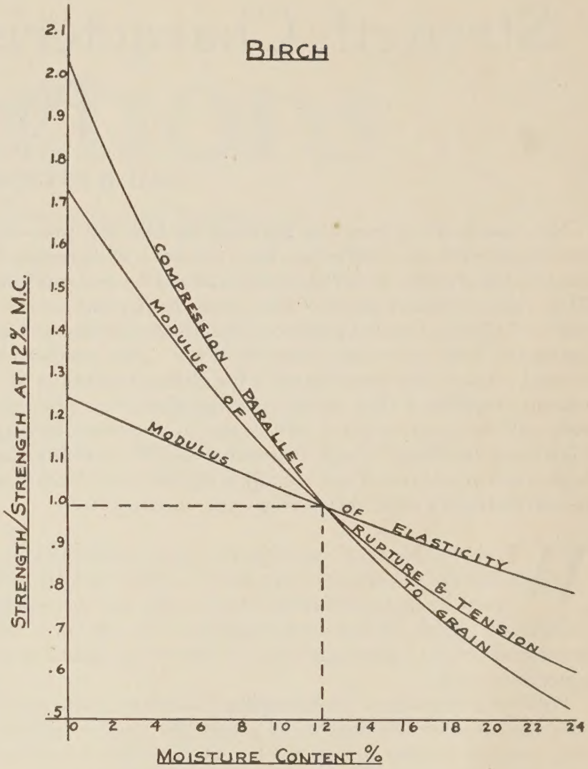

FIG. 1 BIRCH; STRENGTH - MOISTURE CONTENT RELATIONSHIP

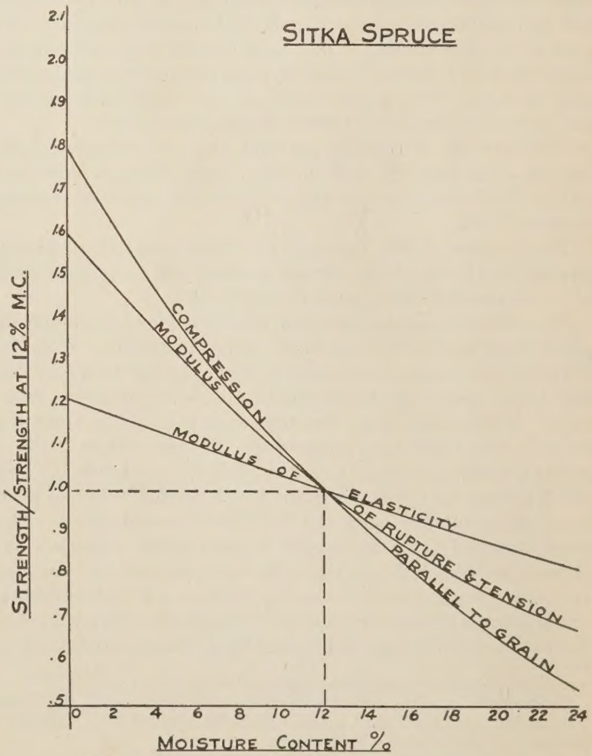

FIG. 2 SITKA SPRUCE; STRENGTH - MOISTURE CONTENT RELATIONSHIP

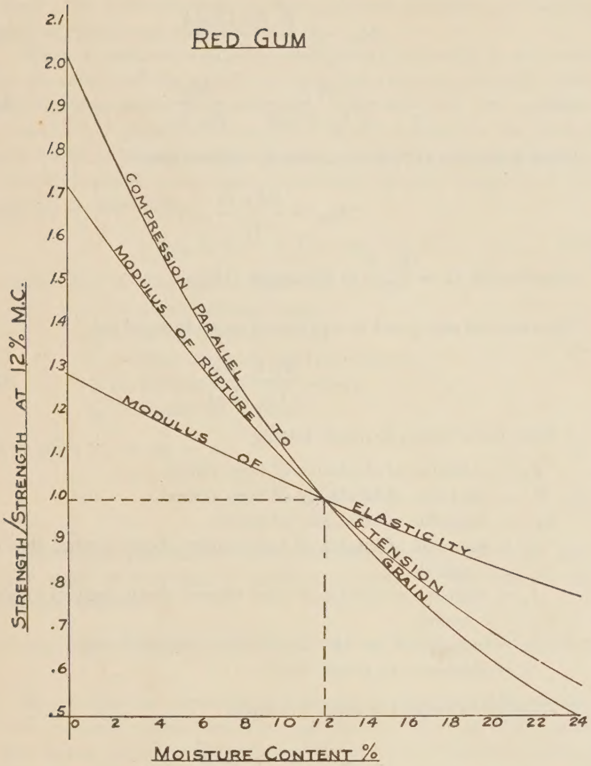

FIG. 3 RED GUM; STRENGTH - MOISTURE CONTENT RELATIONSHIP

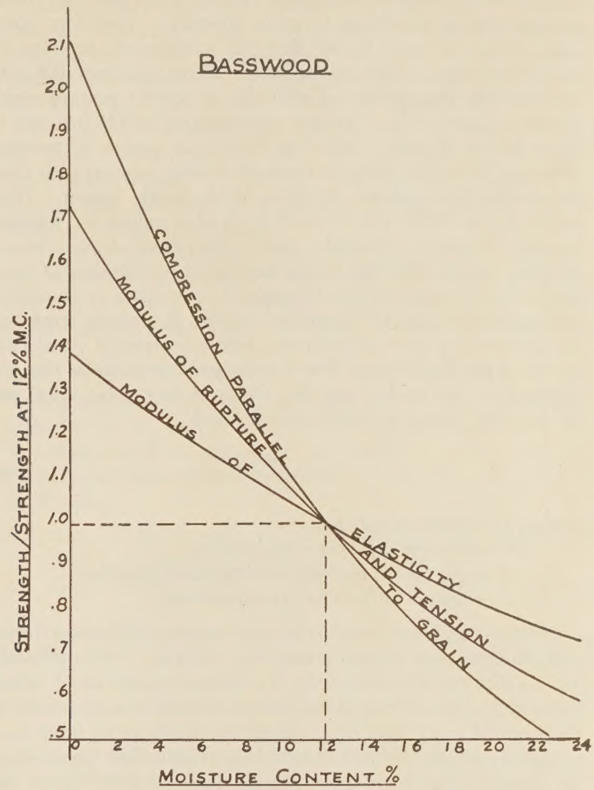

FIG. 4 BASSWOOD; STRENGTH - MOISTURE CONTENT RELATIONSHIP

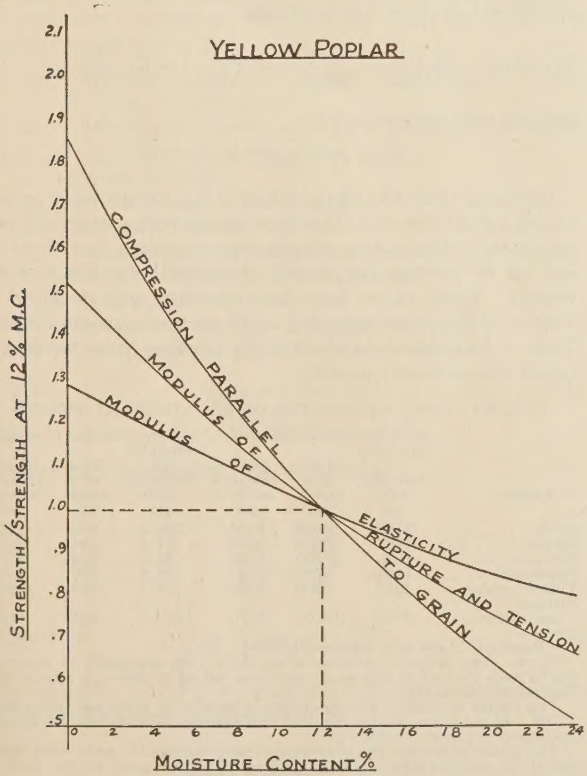

FIG. 5 YELLOW POPLAR; STRENGTH - MOISTURE CONTENT RELATIONSHIP

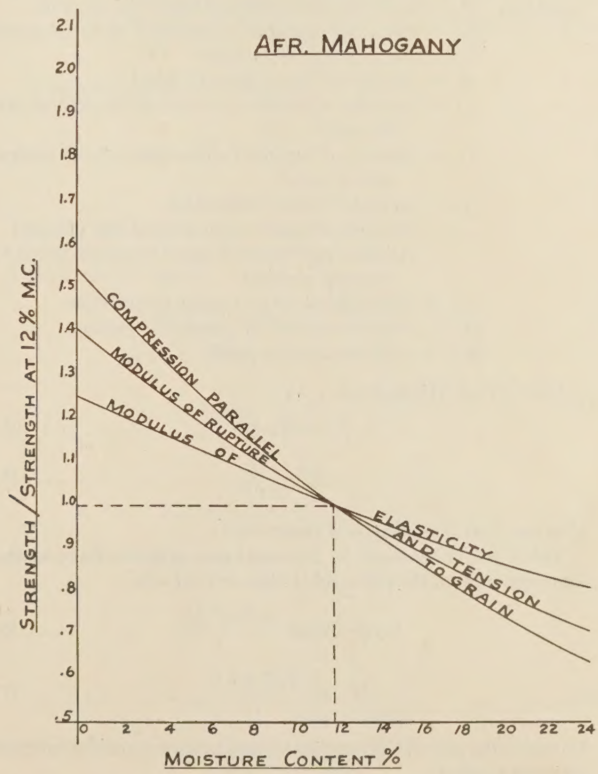

FIG. 6 AFRICAN MAHOGANY; STRENGTH - MOISTURE CONTENT RELATIONSHIP

more difficult problem since each species has a different value parallel and perpendicular to grain direction. Therefore, some means must be found to calculate the modulus of elasticity for any construction of plies and species. There have been few data published on the modulus of elasticity across the grain of wood. However Gassner (4), in a paper presented before the Institute of Aeronautical Sciences, offers an ingenious means of working backward from test data on three-ply panels to determine from these data the apparent modulus of the single veneer. Data published in Table 2:2 of ANC-5 (5) give values for column-bending moduli of three-ply panels determined from column-bending tests on three-ply panels having veneer thicknesses equal and using the same species throughout. In order to determine the apparent moduli for the single veneers, an analysis must first be made of the stress distribution across a plywood panel subjected to bending loads. For a homogeneous material the distribution is well known, and the stress at any point developed by the beam theory is given by the formula

$$f = \frac{MC}{I} \qquad [2]$$

where f = stress at any point
M = moment at section in question
C = distance of fiber from neutral axis of section
I = moment of inertia of entire section

This relationship is based on the assumption of the beam theory that plane sections remain plane after bending. It immediately follows that the stress distribution across a plywood panel, where the modulus of elasticity of the separate veneers changes, is not in the form of a straight line. Since the strain must vary as a straight line from the neutral axis, it is evident that the stress at any point is dependent upon the modulus of elasticity of the veneer in question.

Letting E_V = modulus of elasticity of veneer in question
E_{PI} = apparent modulus of elasticity of entire panel
Δ_V = strain of specific veneer
Δ_P = strain of extreme fiber of panel
I_V = moment of inertia of veneer about neutral axis of panel
I_P = moment of inertia of entire panel about neutral axis of panel
f_V = stress in veneer in question
C = distance of veneer from neutral axis of panel
r = radius of curvature of panel when subjected to bending moment
dA = infinitesimal area at point in question
M_V = moment carried by veneer in question
M_P = total moment on panel

it follows from Hooke's law

$$f_V = E_V \Delta_V \qquad [4]$$

$$\Delta_V = \frac{C}{r} \qquad [5]$$

(See any text on strength of materials.)

This force is balanced by an equal and opposite force at the same distance on the other side of the neutral axis.

$$\text{Force acting} = \frac{E_V C\, dA}{r} \qquad [6]$$

$$M_V = \frac{E_V C\, dA\, C}{r} \qquad [7]$$

But since the sum of the internal moments must equal the external moment applied

$$\Sigma M_V = M_P \qquad [8]$$

$$M_P = \int \frac{E_V C^2\, dA}{r} \qquad [9]$$

and

$$\frac{I}{r} = \frac{M_P}{\int E_V C^2\, dA} = \frac{M_P}{E_{PI}\, I_P} \qquad [10]$$

From Equation [10] it immediately follows that

$$E_{PI} = \frac{\Sigma E_V I_V}{I_P} \qquad [11]$$

substituting $f_V = \dfrac{E_V C}{r}$ in Equation [10].

The stress at any point in a plywood panel is equal to

$$f_V = \frac{M_P C}{I_P} \frac{E_V}{E_{PI}} \qquad [12]$$

Now, for a three-ply panel, letting

E_f = modulus of elasticity of face veneer
E_C = modulus of elasticity of core veneer
E_{PI} = modulus of elasticity of panel
I_f = moment of inertia of face veneer about neutral axis of panel
I_c = moment of inertia of core veneer about neutral axis of panel
I_p = moment of inertia of panel about neutral axis
t = thickness of panel

If all veneers are of the same thickness

$$I_p = \frac{t^3}{12}; \qquad I_c = \frac{t^3}{3^3}/12$$

and $2I_f = I_p - I_c = t^3/12 - t^3/324$

$$\text{Therefore} \quad E_f \left(\frac{t^3}{12} - \frac{t^3}{324}\right) + E_C \left(\frac{t^3}{324}\right) = E_{PI} \frac{t^3}{12} \qquad [13]$$

dividing both sides by $t^3/12$

$$0.963\, E_f + 0.037\, E_C = E_{PI} \qquad [14]$$

Obtaining from ANC-5 (5) values of E_{pI} for three-ply panels, having a grain direction of the face veneers both parallel and perpendicular to the loading, simultaneous equations may be set up and solved yielding the moduli of elasticity for the separate veneers. These values have been obtained and are given in Table 1 of this paper, corrected for 12 per cent moisture content. These values should be checked for accuracy from values obtained from multi-ply panels.

TABLE 1 BASIC STRENGTHS OF SIX AIRCRAFT WOODS[a]

Species	Parallel to grain: Modulus of elasticity, 1000 psi	Parallel to grain: Tension stress, psi	Parallel to grain: Compression stress, psi	Perpendicular to grain: Modulus of elasticity, 1000 psi	Perpendicular to grain: Tension stress, psi	Perpendicular to grain: Compression stress, psi
Birch	2180	16600	8170	106.5	810	399
Spruce	1367	10200	5610	51.4	384	211
Red gum	1535	11900	5800	56.9	442	215
Basswood	1160	8700	4730	36.8	276	150
Yeilow poplar	1511	9200	5290	54.6	333	191
African mahogany	1335	10700	5670	101.6	855	432

[a] Based on 12 per cent moisture content.

NOTE: The values of modulus of elasticity were calculated by means of the formula evolved in this paper and from values of three-ply panels from Table 2:2 ANC-5 (5).

The values of tension and compression parallel to grain are taken from Table 1 of reference (2) for the first five species. The values for African mahogany are from reference (3).

The values of tension and compression perpendicular to grain were determined by multiplying the values for parallel to the grain by the rates of E_T/E_L.

Having the modulus of elasticity of each veneer in a composite panel, the determination of the apparent modulus for bending characteristics can readily be made.

When a member composed of different materials is subjected to an axial load the strain in the panel must of necessity equal the strain in each component part. Also the total force required to strain the panel to this position must be equal to the total of the forces in the component parts. Considering for simplicity a three-ply panel composed of the same species veneers, it can readily be shown that

$$E_{PA}\Delta_P t_p = t_f E_f \Delta_f + t_c E_c \Delta_c \qquad [15]$$

where E_{PA} = modulus of elasticity of panel
Δ_f = strain of face veneers
Δ_c = strain of core veneer
t_f = total thickness of face veneers
t_c = thickness of core veneer
t_p = thickness of panel

but since $\Delta_p = \Delta_f = \Delta_c$

$$E_{PA} = \frac{t_f}{t_p} E_f + \frac{t_c}{t_p} E_c \qquad [16]$$

It has thus been shown that when using a material of this type care must be exercised in choosing the correct modulus for the loading applied.

Allowable Tension and Compression Values of Plywood Panels

Employing the same obvious reasoning that the deflection of each veneer in a panel must be equal to the deflection of the panel and assuming elastic properties of veneers up to failure, a method of least work may be applied to a panel to determine allowable tension and compression stresses for the separate veneers.

Assuming first that all veneers in the panel are of the same species and letting

Δ_L = deflection of veneers with grain longitudinal or in direction of applied load
Δ_T = deflection of veneers with grain transverse or perpendicular to direction of load
f_P = stress in panel
f_L = stress in longitudinal veneers
f_T = stress in transverse veneers
t_L = total thickness of longitudinal veneers
t_T = total thickness of transverse veneers
t_P = total thickness of panel
E_L = modulus of elasticity of longitudinal veneers
E_T = modulus of elasticity of transverse veneers

Then $$f_P t_P = f_L t_L + f_T t_T \qquad [17]$$

but $f_L = \Delta_L E_L$ and $f_T = \Delta_T E_T$

solving for a relationship between f_L and f_T since

$$\Delta_L = \Delta_T$$

$$\frac{f_L}{f_T} = \frac{E_L}{E_T}$$

Therefore $$f_p t_p = f_L t_L + f_L \frac{E_T}{E_L} t_T \qquad [18]$$

For simplification, if X = percentage of longitudinal veneers, and Y = percentage of transverse veneers

$$100 f_P = X f_L + Y \frac{E_T}{E_L} f_L \qquad [19]$$

It can safely be assumed then that the longitudinal veneers fail first and, therefore, using the modulus of rupture and maximum crushing strength of the longitudinal veneers the tension and compression allowables for a panel may be calculated. These allowable values for the six woods described are given in Table 1.

The assumption that elastic qualities hold up to failure should not cause too great an error, since the elastic limit of wood is quite close to the ultimate. Also the ratio of E_L/E_T appears to be greater than the ratio of modulus of rupture longitudinally to modulus of rupture transversely.

If a panel is so constructed that the longitudinal veneers are of different species the strength of the panel is evidently dependent upon the strength of the weaker species and the allowable strength of the stronger veneer must be reduced so that $f_L = (E_L/E_L{}^I)f_L{}^I$.

Poisson's Ratio for a Multi-Ply Panel

Very few data have been published on the value of Poisson's ratio for various species of wood. In a paper by C. F. Jenkins (6), these values are given for four species of wood. Knowing these basic values Poisson's ratio for a complete panel may be calculated.

Let μ_L = Poisson's ratio for veneers having longitudinal grain in direction of loading
μ_T = Poisson's ratio for veneers having transverse grain in direction of loading
μ_P = Poisson's ratio for entire panel in direction of loading
E_{PP} = modulus of elasticity of panel for axial load in direction of loading
E_{PT} = modulus of elasticity of panel for axial load perpendicular to direction of loading
Δ_{PP} = deflection of panel in direction of loading
Δ_{PT} = deflection of panel perpendicular to loading
A = deflection of longitudinal veneers perpendicular to direction of loading if unrestrained
B = deflection of transverse veneers perpendicular to direction of loading if unrestrained

Considering first a three-ply panel, Fig. 7

$$A = \frac{\mu_L f_L}{E_L}; \quad B = \frac{\mu_T f_T}{E_T}$$

$$\Delta_{PT} - B = \Delta_{LI}$$

$$A - \Delta_{PT} = \Delta_{TI}$$

Forces must be equal

$$XE_T(A - \Delta_{PT}) = YE_L(\Delta_{PT} - B) \qquad [20]$$

$$XE_T A - XE_T\Delta_{PT} = YE_L\Delta_{PT} - YE_L B \qquad [21]$$

$$\Delta_{PT} = \frac{XE_T A + YE_L B}{YE_L + XE_T} \qquad [22]$$

$$\Delta_{PT} = \frac{X\mu_L f_L \frac{E_T}{E_L} + Y\mu_T \frac{E_L}{E_T} f_T}{YE_L + XE_T} \qquad [23]$$

but by Maxwell's theorem

$$\frac{\mu_L}{E_L} = \frac{\mu_T}{E_T}$$

$$\Delta_{PT} = \frac{\mu_L f_L \frac{E_T}{E_L}(X + Y)}{YE_L + XE_T} \qquad [24]$$

$$\Delta_{PP} = \frac{Xf_L + Yf_T}{100E_{PP}}$$

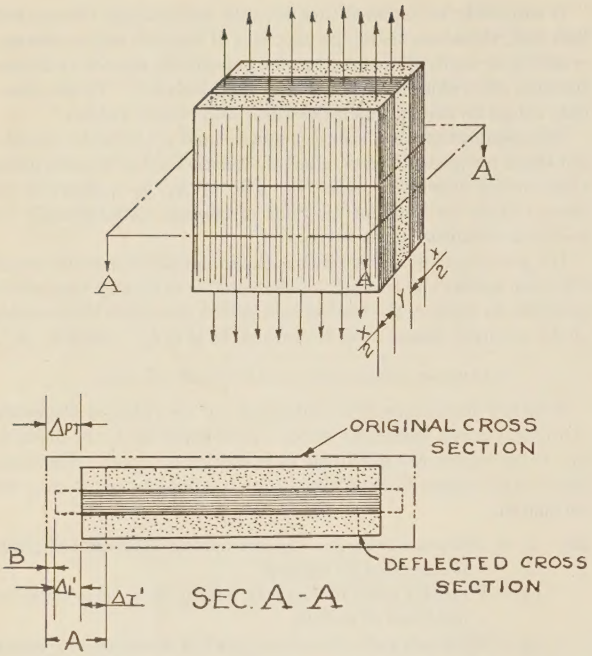

Fig. 7 Construction of Three-Ply Panel

$$YE_L + XE_T = 100E_{PT}$$

and $X + Y = 100$

solving
$$\mu_P = \frac{100\mu_L \frac{E_T}{E_L} E_{PP}}{\left(X + Y \frac{E_T}{E_L}\right) E_{PT}} \qquad [25]$$

This proof is again based on the assumption that the panel is composed of veneers of the same species throughout and, as before, if dissimilar veneers are used, a correction factor depending upon the ratio of the moduli of elasticity of the two materials must be introduced.

Knowing the bending modulus of elasticity of the panel and the value of Poisson's ratio, it is felt that the shear modulus may be calculated from the formula found in any standard text on strength of materials

$$G = \frac{E}{2(I + \mu)} \qquad [26]$$

Using this relationship and basic values of μ_L obtained from Jenkin's report (6) the value of modulus of rigidity of a three-ply spruce panel checks fairly accurately with values suggested in ANC-5 (5).

Concentrated Loads on Plywood Panels

Probably the main disadvantage up to this time in wooden structures for aircraft use has been the question of attaching fittings which involve high concentrations at the point of attachment. An excellent example is the connection required to attach an outer panel or a complete wing to a center-section panel or fuselage. The magnitude of the loads involved has necessitated a large number of bolts spread out over a considerable distance because of the low bearing value of bolts in wood. The introduction of these bolt holes has greatly reduced the efficiency of the wood at this point, and the bolts required have greatly increased the weight of the construction.

The idea suggested by the Forest Products Laboratory, Madison, Wis., of introducing at these critical sections impregnated-compregnated wood seemed a practical solution. This process requires the impregnation of wood locally with a urea-formaldehyde solution, and the application of rather high pressures to obtain a very dense material. This wood then has very high bearing values and is impervious to moisture-content changes. The difficulties encountered thus far in the application of this process have been the high pressures required and the control of the amount of impregnation.

With the development of a new adhesive, capable of gluing metal to metal or metal to wood, this problem of local concentrations seems to have been solved. One of the advantages of this new adhesive is the fact that it requires the same cycle as presently used in the bag-molding process, making it possible to introduce locally a plate having high bearing strength sandwiched between the veneers of the panel. Tests run on this adhesive have given almost unbelievable results. At room temperature, a shear allowable of 2000 psi and a tension value of 1000 psi are very conservative.

The trend in the plastic-bonded-plywood field has been toward true monocoque construction, requiring the shell to carry all loads. In such a construction the introduction of necessary access holes and cutouts for such items as landing lights causes concentration of stresses around these sections, which concentrations are far more critical than in the conventional skin-stringer construction. The use of this new glue makes it possible to introduce into the panel around these holes a plate of aluminum or some other material to act as a doubler plate. These plates can be placed in position during the lay-up of the panel, and the necessary holes cut through later. The only requirement evident at present is that the plate introduced should have the edges tapered gradually so that a not too abrupt change in section is obtained.

If the metal plate is placed on the center line of the plywood panel, or if two plates are glued at similar distances from the center line so that balanced construction remains, there will be no warping evidenced. However, if a metal plate be glued to one side of a panel, the moment introduced by the difference in thermal contractions upon cooling of the panel and plate, and the radius of curvature of the warping can be calculated quite easily if the coefficients of expansion of the materials are known.

Conclusion

The author has prepared Table 2 to compare the basic-strength

TABLE 2 RELATIVE STRENGTHS OF VARIOUS MATERIALS

Material	Specific gravity	Thickness, in.	Allowable loads: Tension, lb per in. width	Block compression, lb per in. width	Shear, lb per in. width	EI
24 ST Al. alloy	2.77	0.052	3200	3200	1924	120.7
24 SRT Al. alloy	2.77	0.052	3380	3380	2030	120.7
1025 Steel	7.85	0.019	1045	1045	665	16.0
4130 Steel	7.85	0.019	1810	1810	1045	16.6
Magnesium alloy	1.80	0.080	2560	2560	1530	277.3
Spruce	0.40	0.358	3650	2010	412	5230.0
Plywood:						
Birch face, Poplar core	0.66	0.205	1918	1304	460[a]	1488.6
Mahog. face, Poplar core	0.58	0.247	1738	956	550[a]	1598.4

[a] Estimated.

Note: The plywood panels are constructed of either birch or African mahogany face veneers of 1/20 and longitudinal grain direction, and having five and seven, respectively, yellow poplar veneers of 1/48 between the faces. These poplar veneers have the outside veneer longitudinal, and then successive veneers have grain directions at right angles.

values of plastic-bonded plywood with those of other materials popular in aircraft manufacture. The table is based upon unit width of material and is so arranged that the specific gravity multiplied by the thickness remains a constant. At first glance it appears that plywood is quite far down the list in strength characteristics but this is not necessarily the case. In aircraft construction, the design criterion is usually one of buckling which is dependent upon the EI or stiffness value of the panel under consideration. It its because of this buckling phenomenon that in thin-metal construction, stringers and stiffeners must be added throughout to increase the effective width of sheet capable of carrying compressive loads. Although this addition of stringers and stiffeners strengthens the sheet and makes it capable of carrying the load applied, anyone who has ever done any flying will vouch for the fact that wrinkles are still very much in evidence. To avoid these wrinkles which are evidently harmful to airfoil shapes and laminar-flow characteristics requires a construction in thin metal which is heavy, costly, and quite impractical. Plywood on the other hand having such a high value of stiffness factor EI and allowing the use of a much thicker section has no tendency toward wrinkling and can with comparative ease be designed to prevent buckling from occurring.

It has been demonstrated in a series of tests conducted by the author's company that plywood has the ability to carry appreciable loads after buckling. Since all airplanes are designed to carry a 50 per cent overload, it is possible to design plywood wings to avoid buckling up to the loading of any normal flight condition and to allow buckling to occur safely above this load. This factor, combined with the fact that there are no rivet heads or skin laps to interfere with air flow, greatly enhances its application in high-speed airplanes. Since the elastic limit is very near the ultimate in wood, the advantage of having elastic deformation for any normal flight loads is apparent.

Contrary to public opinion, plywood fabricated with the resinous products used today has a greater resistance to fire than has an equal weight aluminum-alloy part. This is especially true when in the presence of flaming oil or gasoline.

The added advantage of having bullets leave clean holes in plywood and the ability to repair with ease such holes makes this material especially attractive in the present crisis.

These are but a few of the many advantages of this material. Designers are today thinking of its application as a substitute for the now precious metals but are discovering in its application the many advantages to be obtained from its low cost, ease of fabrication, and good strength characteristics.

It is firmly believed that with gluing technique advancing at so rapid a pace it will not be in the too distant future that the application of a thin metal covering glued to a plywood core will evidence itself in airplanes having extremely high wing loadings. This construction—long a dream of aircraft designers—will combine the high axial strengths of the metal with the good buckling characteristics of the thicker plywood constructions.

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Bearing Strength of Plastics and Plywood

By JAMES BOND,[1] LAWRENCE, KAN.

This paper reports the results of tests carried out under the supervision of Prof. Earl D. Hay in the materials-testing laboratory of the University of Kansas to determine the bearing strength of commercially available high-impact molded and laminated plastic sheets and resin-bonded birch plywood. Curves are shown for load versus deformation in per cent of bearing-pin diameter, and load versus permanent set in per cent of bearing-pin diameter. The tests demonstrated the inferiority of plywoods in bearing strength as compared with the reinforced plastics. Heating in most cases improves the bearing strength of specimens, while soaking them in water definitely decreases their strength, as compared with results on specimens conditioned at room temperature and a relative humidity of 55 per cent.

PLASTICS are being used today in many and varied applications, but as yet few actual design data as to their physical characteristics are available. It was with this in mind that the development of a method for determining the bearing strength of several reinforced plastics was undertaken. The test methods discussed in this paper are applicable to all types of plastics.

Materials and Apparatus

The test specimens used in these tests were prepared from sheets of six different types of commercially available plastics, namely, laminated-canvas phenolic plastic $^1/_2$ in. thick; molded macerated-filled phenolic plastic $^9/_{16}$ in. thick; laminated-paper phenolic plastic, grade XX, $^1/_2$ in. thick; 3-ply birch plywood 0.11 in. thick made with $^1/_{24}$-in-thick veneers; 7-ply birch plywood 0.14 in. thick made with $^1/_{48}$-in-thick veneers; and 27-ply birch plywood $^1/_2$ in. thick made with $^1/_{48}$-in-thick veneers. The plywoods were bonded with phenolic-resin film. Henceforth in this paper these materials will be referred to as canvas, macerated, XX, 3-ply, 7-ply, and 27-ply, respectively.

The specimens were cut to size with an ordinary hollow-ground circular saw and the bearing hole, which was $^3/_8$ in. diam, was drilled and reamed to size. Care was taken to avoid charring the material while drilling in order that true results could be obtained from the tests. The plywood specimens were cut to a size of $1^1/_4$ in. $\times$ $1^1/_8$ in. and, in the case of the 3-ply and 7-ply specimens, enough sheets were stacked together to give an approximate thickness of $^1/_2$ inch. The other three specimens were cut to a size of $1^3/_8$ in. $\times$ $1^1/_8$ in. This slight increase in size was made in order to be sure that the specimen would fail in bearing rather than in some other manner.

The specimens were conditioned in three different ways. One set of specimens was placed in a desiccator over a saturated solution of calcium nitrate. This solution maintained a relative humidity of approximately 55 per cent in the desiccator. Another set of specimens was immersed in water at a room temperature of approximately 70 F. The third set of specimens was placed in a drying oven which was maintained at a temperature of 160 F. All specimens were allowed to remain in their conditioning medium for a period of at least 96 hours before any testing was done. The specimens were tested immediately upon removal from the conditioning medium. The tests required approximately 10 to 15 minutes for completion.

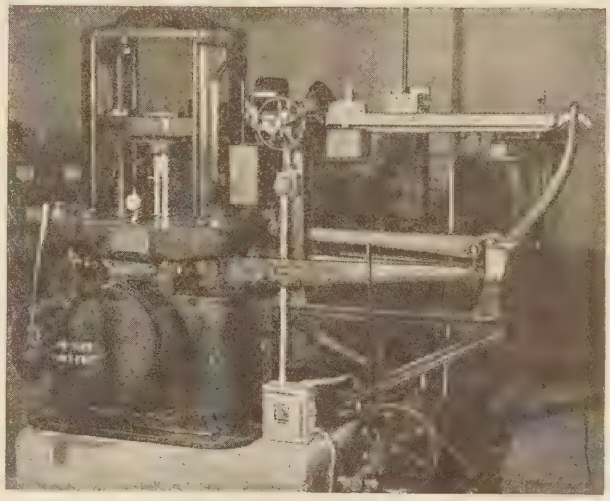

Fig. 1

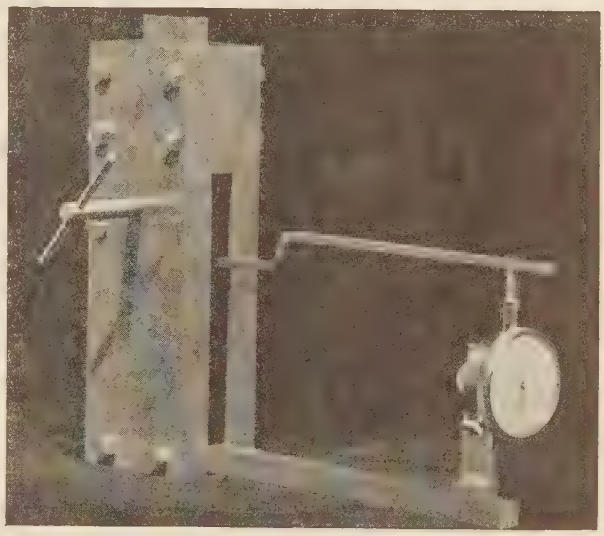

Fig. 2

The testing was done in a Riehle 40,000-lb testing machine, Fig. 1, at room temperature, and the loads were applied and removed at a rate of 0.0529 in. per min. The specimens were placed in a jig, Fig. 2, which is similar to the one proposed for this type of testing by the Glenn L. Martin Company. This jig consists essentially of two heavy metal plates separated by two spacer blocks. A plunger slides between the spacer blocks and impinges upon the top of the test specimen. This test specimen

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Contributed by the Rubber and Plastics Subdivision of the Process Industries Division and presented at the Semi-Annual Meeting, Cleveland, Ohio, June 8–10, 1942, of The American Society of Mechanical Engineers.

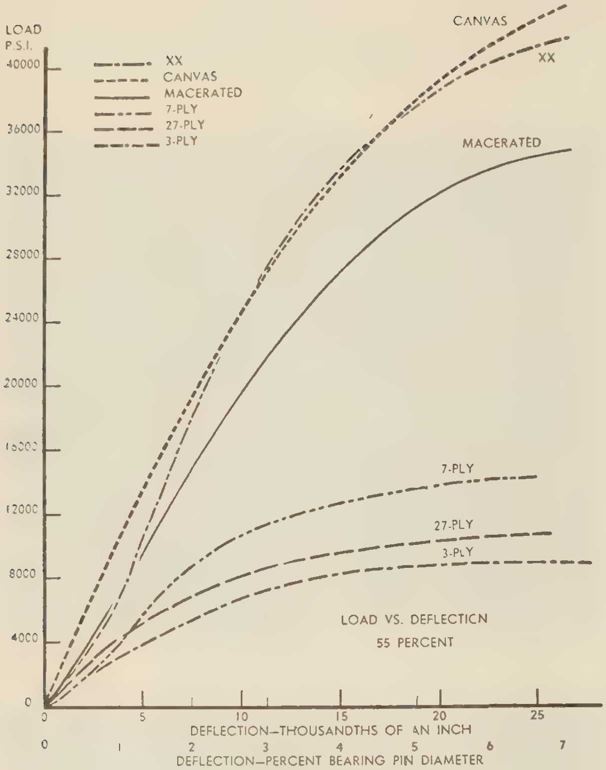

Fig. 3

is held in place by a bearing pin which runs through the side plates and the specimen, and is constructed of $^3/_8$-in. ground and polished drill rod. In this jig, the part of the specimen above the bearing pin is compression-loaded when a force is applied to the plunger. The part of the specimen below the bearing pin is essentially unstressed and the deformation, attributable to bearing pressure is measured at this point through the use of a multiplying lever. The multiplying lever has two purposes; i.e., it places the Ames dial gage in a more convenient place for reading, and it also multiplies the deformation so that less error will be made in reading small deformations. The multiplying lever used in these tests was 5 in. long. By means of this lever readings may be made directly to 0.0002 in. Deflections were not measured until the rapid initial deformation had occurred in the test specimens (0.5 to 1 min). The test jig was accurately centered between the crossheads of the testing machine so that the load would not be applied eccentrically.

Load Versus Deformation Test

The United States Army Air Corps has proposed a test to determine the bearing strength of plastic materials.[2] In this test, the maximum allowable bearing load is to be that load which causes a deformation in the plastic equal to 4 per cent of the diameter of the bearing pin. This test can be performed quite easily with the jig previously described. A load is placed on the specimen and the deformation is noted when the machine is balanced. A series of readings was taken from zero loading up to a loading which gave a deformation in the specimen equal to approximately 7 per cent of the bearing-pin diameter.

The test was run using specimens prepared as previously noted.

[2] "Structural Plastics in Aviation," by J. B. Johnson, *Modern Plastics*, Nov., 1941, p. 214.

TABLE 1 BEARING STRENGTH TESTS—LOAD VERSUS DEFORMATION

Material	Load causing a deformation equal to 4 per cent of bearing-pin diameter on specimens conditioned for 96 hr at 160 F, psi	70 F and 55 per cent relative humidity, psi	70 F in water, psi
Canvas	32900	32900	26300
Macerated	31800	26900	25900
XX	36200	33100	26600
27-Ply	15400	9500	4500
7-Ply	16500	12500	5600
3-Ply	9100	8100	3800

The tabulated results for the bearing load at the loading which gave a deformation equal to 4 per cent of the diameter of the bearing pin appear in Table 1. Eight specimens of each material were tested for the 55 per cent conditioning treatment, and the data were averaged to obtain the curves shown in Fig. 3. Four specimens were used for the conditioning treatment at 160 F, and for the water immersion, and the data averaged to obtain the curves shown in Figs. 4 and 5.

Load Versus Permanent Set Test

A test for determining the bearing strength of plastics has been proposed in which the maximum allowable bearing load is to be that load which causes a permanent set in the material equal to 0.2 per cent of the bearing-pin diameter.[3] For plastic materials which do not have a well-defined yield point, an arbitrary index of elastic strength must be chosen which takes into consideration the amount of permanent deformation a material can have with-

[3] "Proposed Specifications for Structural Thermosetting Plastics," Glenn L. Martin Co. Engineering Report No. 1432, April, 1941.

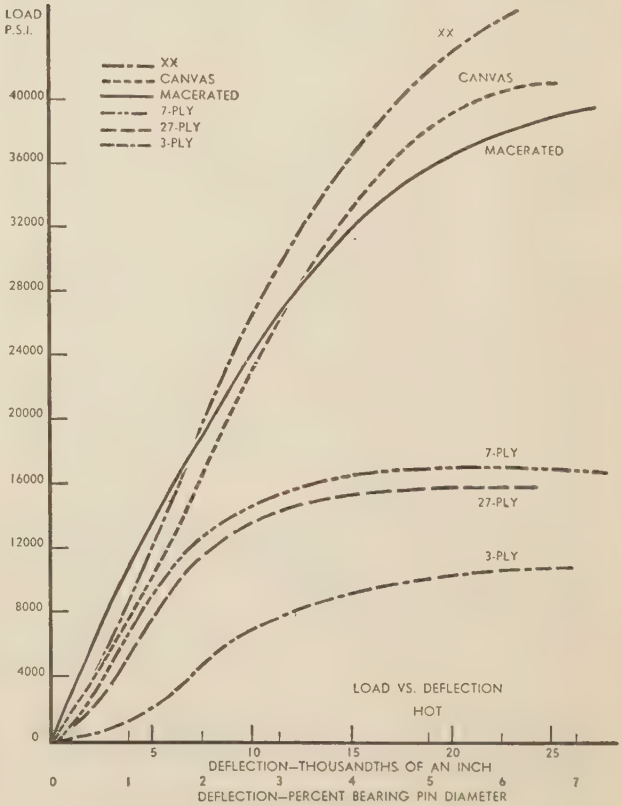

Fig. 4

out appreciable structural damage. For other materials not having a well-defined yield point, a limiting set of 0.2 per cent has been found satisfactory for locating the yield strength. Therefore it seemed logical to assume a limiting value of 0.2 per cent permanent set in determining the applied bearing stress of structural thermosetting plastics.

In this test, the specimen was loaded to about 10 per cent of the maximum bearing load as determined by a preliminary test, the load and deformation were recorded, and the load was then removed at the same rate at which it was applied until zero loading was reached. The deformation at zero loading was recorded as the permanent set for that loading. A series of these tests was run from zero loading up to a loading which gave a permanent set to the specimen of approximately 2 per cent of the diameter of the bearing pin. The test specimens were conditioned as previously noted. The tabulated results for the bearing load at that loading which causes a permanent set equal to 0.2 per cent of the diameter of the bearing pin are indicated in Table 2. Average curves of load versus permanent set were plotted for all six types of material conditioned in the three different ways, using 4 to 8 specimens for each condition as in the deformation tests. They appear in Figs. 6 to 8, inclusive.

TABLE 2 BEARING STRENGTH TESTS—LOAD VERSUS PERMANENT SET; 0.2 PER CENT

Material	Load causing a permanent set equal to 0.2 per cent of bearing-pin diameter on specimens conditioned for 96 hr at— 160 F psi	70 F and 55 per cent relative humidity, psi	70 F in water, psi
Canvas	4100	8100	4200
Macerated	5700	10700	6900
XX	12600	3100	9300
27-Ply	1000	1900	300
7-Ply	2100	500	1100
3-Ply	600	700	500

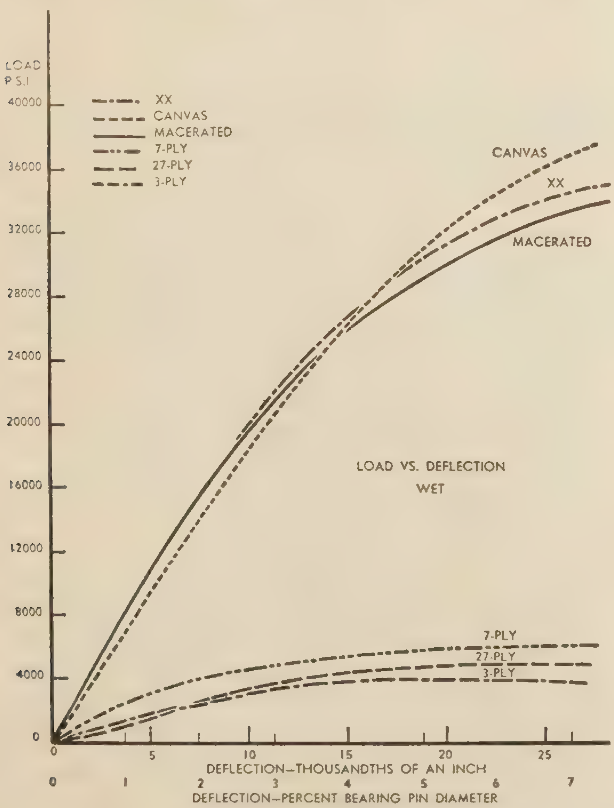

Fig. 5

Discussion of Results

The unit bearing pressure for both tests was determined from the formula $S_b = P/1.57rl$, in which S_b equals the unit bearing pressure in pounds per unit area, P equals the applied force in pounds, r equals the radius of the bearing hole, and l equals the thickness of the specimen.[4] The formula is evolved assuming that the radial pressure distribution varies as a cosine curve with the pressure equal to zero at the sides of the bearing hole and a maximum at the center. If the radial pressure had been assumed constant, the formula would have become $S_b = P/2rl$. This last equation is used in practice because of its simplicity, but the first equation represents a better approximation of the true unit bearing pressure.

Load Versus Deformation. In the test as proposed by the Army Air Corps, heating at 160 F for 96 hr increased the bearing strength of the specimens and soaking in water at 70 F for 96 hr decreased their bearing strength, as compared to the strength of those specimens conditioned at room temperature and 55 per cent relative humidity. The bearing strength in this test is equal to the loading required to cause a deformation in the specimen equal to 4 per cent of the diameter of the bearing pin.

The canvas and XX products had the greatest bearing strength of the materials tested and were followed closely by the macerated material. The plywoods were definitely inferior to the reinforced plastics of higher resin content. The bearing strength of the plywoods in the dry condition (heated at 160 F) was approximately 3 times as great as that observed for the same materials in the wet condition. The 7-ply product had the highest bearing strength of the group of plywoods. Its superiority to the 27-ply product, which was made with the same thickness of veneer ($^1/_{48}$ in.) and same resin film, can possibly be attributed to less complete curing of the resin in the thicker material. The lower bearing strength of the 3-ply product can be explained on the basis of the lower resin content of this plywood which was made with $^1/_{24}$-in. veneer.

Load Versus Permanent Set. In the load versus permanent set test, the proposed limiting value of permanent set, 0.2 per cent of the pin diameter, amounted to only 0.00075 in. or a dial reading of 3.75 divisions. This means that, in order for a test to be run with any degree of accuracy, the gage which measures deflection must have no lost motion. The type of dial gage used gives accurate readings when the dial readings are increasing; but when the readings start to decrease they cannot be assumed to be entirely accurate because of the inherent lost motion in reversing the direction. It would seem logical to assume that there is considerable error in determining a permanent set as small as 0.2 per cent because of failure of both instrument and specimens to respond in a reproducible way. Therefore, it is believed that no conclusions can be drawn from the erratic results shown in Table 2.

TABLE 3 BEARING STRENGTH TESTS—LOAD VERSUS PERMANENT SET; 1.6 PER CENT

Material	Load causing a permanent set equal to 1.6 per cent of bearing-pin diameter on specimens conditioned for 96 hr at— 160 F, psi	70 F and 55 per cent relative humidity, psi	70 F in water, psi
Canvas	40000	42300	34500
Macerated	34500	35200	28600
XX	42600	39000	34200
27-Ply	13800	8800	4800
7-Ply	14700	9400	5200
3-Ply	8500	6400	3900

In order to get allowable unit pressures, comparable to those as found by the load versus deformation test, permanent sets up to 3 per cent of the bearing-pin diameter were encountered with

[4] "Strength of Materials," by N. C. Riggs and M. M. Frocht, Ronald Press, 1938, p. 55.

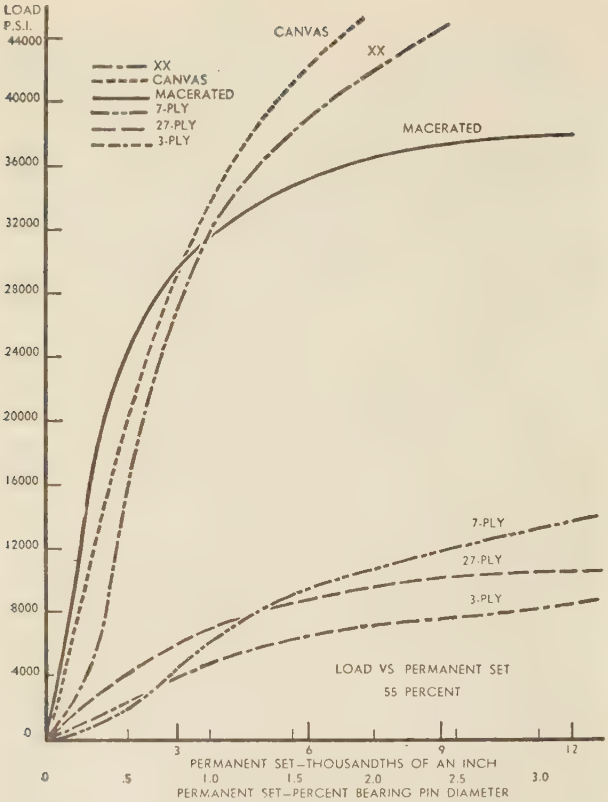

Fig. 6

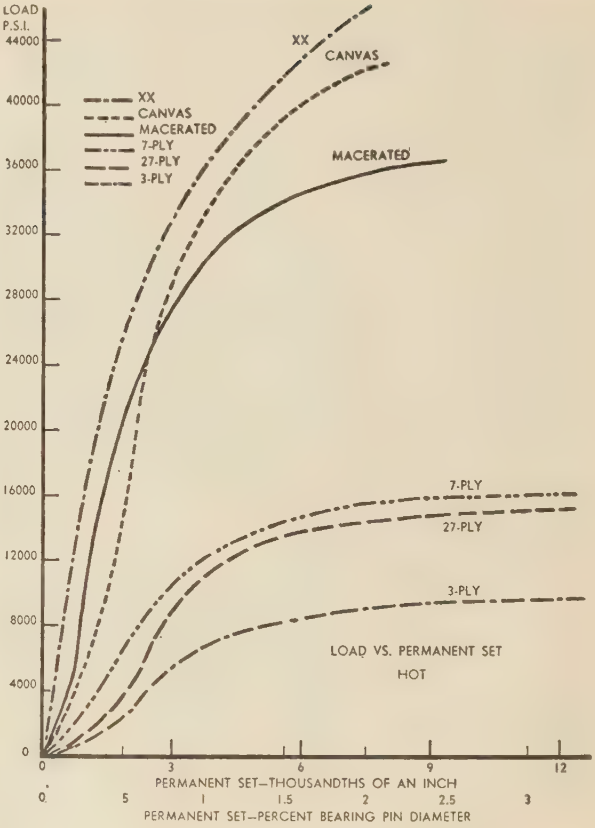

Fig. 7

an average value of about 1.6 per cent. Bearing loads which give a permanent set of 1.6 per cent have been read from the curves in Figs. 6 to 8 and are shown in Table 3. These data are in agreement with the conclusions derived from the deformation data.

Summary

The specimens used in these tests were representative of commercially available high-impact molded and laminated plastic sheets and resin-bonded birch plywood. The testing was done in a jig similar to that proposed by the Glenn L. Martin Company, for making bearing tests on sheet plastics. Two main types of test were run, i.e., load versus deformation and load versus permanent set. Curves were plotted for load versus deformation in per cent of bearing-pin diameter, and load versus permanent set in per cent of bearing-pin diameter.

Three sets of test specimens were used: One set was conditioned in a desiccator at 55 per cent relative humidity for 96 hr; one set was immersed in water at room temperature of approximately 70 F for 96 hr; and one set was placed in a drying oven in which the temperature was maintained at approximately 160 F for 96 hr. The main tests were done at 55 per cent relative humidity, and the other two tests were run to determine the effect of extremes in temperature and moisture upon the bearing strength of the specimen.

The plywoods were definitely inferior to the reinforced plastics in bearing strength. In nearly all cases heating improved the bearing strength of the specimens and soaking in water decreased their bearing strength, as compared to that observed for the specimens conditioned at room temperature and a relative humidity of 55 per cent.

The proposed load versus deformation (4 per cent of bearing-pin diameter) test gave much more satisfactory results than did the proposed load versus permanent set (0.2 per cent of bearing-pin diameter) test and, in addition, was easier to perform. For these two reasons the former test is the more desirable.

Acknowledgments

This project was carried out in the materials testing laboratory of the University of Kansas, under the supervision of Prof. Earl D. Hay, head of the department of mechanical engineering. The problem was suggested by Dr. G. M. Kline, National Bureau of Standards, and the materials supplied directly by him were obtained through the courtesy of the following firms:

The Bakelite Corporation supplied molded macerated-fabric-filled phenolic compound, BM-199 Tan.

Continental-Diamond Fibre Company supplied laminated-canvas phenolic plastic, Grade C Natural.

Resinous Products and Chemical Company supplied birch plywoods, bonded at 300 F, with Tego resin film No. 2.

Discussion

W. N. Findley.[5] In the load-deformation diagrams which Mr. Bond has shown, Figs. 3, 4, and 5 of the paper, he noted a curvature at the initial part of the load-deformation diagram followed by a straight-line portion and a final curved region. The initial curved portion which he observed was not uniform in all tests but was sometimes concave upward and sometimes concave downward.

This suggests that perhaps a lag occurred in the instrument

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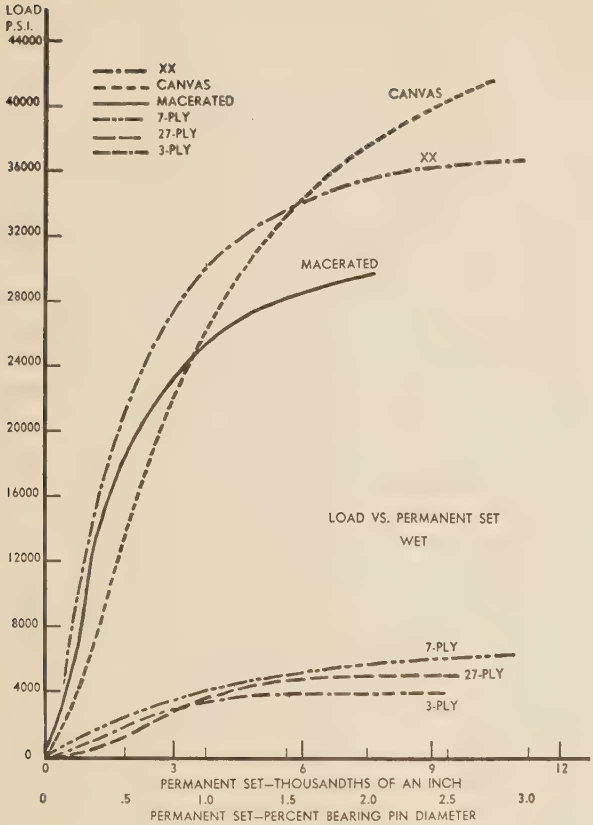

FIG. 8

which might have affected the results. Multiplying-lever-type of extensometers and deformation gages, such as those which Mr. Bond used, are frequently subject to an initial lag. This lag is the result of elastic deformation of the instrument under the frictional forces involved in its movement. Backlash, if present, may also contribute to this lag, or it may also produce the opposite effect to a lag depending upon the initial conditions of the instrument. As a result of this effect the load-deformation curve plotted from the uncorrected data will usually not go through the origin even for materials as truly elastic as spring steel, and no attempt should be made to force the curve through the origin.

If the foregoing should be the cause of the shape of curve which the author found, then it would be possible that the values of bearing strength which he obtained from the load, corresponding to a deformation equal to 4 per cent of the bearing-pin diameter, might be seriously in error.

An examination of the load versus permanent set curves, Figs. 6, 7, and 8 of the paper shows a characteristic similar to that for the load-deflection curves. This also suggests that instrument lag may have influenced the results. If the curvature at the foot of the diagrams in Figs. 6, 7, and 8, has arisen from this cause, one method of avoiding the trouble would be to measure the set at a certain positive value of load, rather than at zero load. It would then be possible, when decreasing the load, to back off below the "set" load, then increase to the "set" load. This might avoid some if not all of the difficulty from backlash.

A third method called the "offset" method has two advantages to recommend it. This method determines the load at which the load-deformation curve deviates from a straight line by a specified amount[6] rather than the load which will produce a given deformation, as in the "4 per cent deformation" method. It requires much less time to perform than the permanent-set method, while yielding just as reliable results.

In the offset method a load-deformation test is run and a load-deformation diagram plotted. A smooth curve is then drawn through the plotted points and extrapolated downward, as

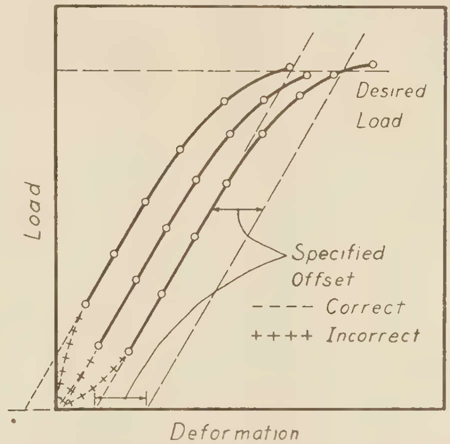

FIG. 9 POSSIBLE LOAD-DEFORMATION DIAGRAMS OBTAINED WITH MULTIPLYING-LEVER-TYPE DEFORMATION GAGES

a straight line, until it cuts the axis of zero load, see Fig. 9 of this discussion. Then the specified offset is measured from this point of intersection, and a straight line is drawn parallel to the straight-line portion (or foot) of the curve and at the desired offset; see Fig. 9 of this discussion. The intersection of this line with the load-deflection curve is the desired bearing load.

In order to eliminate possible variations between specimens, the author averaged the data from several specimens and then plotted the result. The writer feels that this practice tends to obscure the trend and characteristics of the data. A better practice might be to obtain the bearing strength from each test separately and then average these values of bearing strength. This method also makes the detection of errors of observation much easier.

Since the bearing-strength test is designed primarily to evaluate the ability of a material to resist the bearing stress encountered in riveted joints, rather than to evaluate a basic property of the material, the writer questions the advisability of using the equation ($S_b = P/1.57rl$) which the author used in calculating his bearing strengths. If the bearing strength calculated by this equation were used in the design of a riveted joint by the usual method, the bearing strength of the joint would be about 30 per cent weaker than expected. This would be due to the fact that the equation usually used is ($S_b = P/2rl$) instead of the equation as given by the author.

The actual distribution of stress in a riveted joint is uncertain because, in such a joint, there is usually not a close fit between the shank of the rivets and the holes in the plates. Thus, it may be argued that attempts to calculate anything but an average stress on an individual rivet are of doubtful value.

The writer has applied the offset method to the author's load-deformation curves Figs. 3, 4, and 5. The results for

[6] This method is widely used to determine the yield strength of materials in tension, compression, and torsion tests; see "Standard Definitions of Terms Relating to Methods of Testing," designation E 6–36, American Society for Testing Materials, "Book of Standards" 1939, part 1, p. 779.

TABLE 4 RESULTS OF APPLYING OFFSET METHOD

Bearing strength for 0.2 per cent offset ($S_b = P/2rl$, psi)

Material	Hot	55 Per cent	Wet
Canvas	23500	...	20600
Macerated	23000	18400	...
X X	24600	21600	...
27 Ply	...	6400	...
7 Ply	...	7500	...
3 Ply	5500	5500	...

Bearing strength for 1.6 per cent offset ($S_b = P/2rl$, psi)

Material	Hot	55 Per cent	Wet
Canvas	31700	...	28800
Macerated	29600	26200	...
X X	35000	30600	...
27 Ply	12100	7900	...
7 Ply	12800	10200	...
3 Ply	7800	6800	...

values of offset of 0.2 per cent and 1.6 per cent are shown in Table 4 of this discussion for all of the author's curves for which the data were sufficiently conclusive. The fact that the data for these curves were averages of several tests, together with the fact that the plotted points were not shown, made determination of the bearing strength by the offset method impossible in some cases. The values shown were computed from the usual equation ($S_b = P/2rl$). It will be noticed that this method yields consistent results for values of offset as small as 0.2 per cent. However, the writer feels that larger values of offset such as one or two per cent would be more useful for plastics.

Mr. Bond has reported bearing tests of plastics which were subjected to three different conditioning procedures. It may be of interest to mention here the effect of such conditioning procedures on the compression strength of plastics as determined at the University of Illinois. In these tests, three groups of specimens from the same material were conditioned in three different ways similar to the procedures used by the author and then stored in a constant-temperature, constant-relative-humidity atmosphere. Specimens from each group were then tested in compression at successively longer intervals of time following the conditioning. The data show that a period of time of about one month was required for the weight and strength of the plastics tested to return to the value which obtained before the material was subjected to the conditioning procedure. Thus it would appear that very precise test results might be difficult to obtain without rather long periods of conditioning at constant temperature and relative humidity.

Author's Closure

The author is in agreement with Mr. Findley as to the possibility of an initial lag or backlash being present in the measuring instrument. This fact was pointed out in the paper. It is entirely possible that these factors affected the results of these tests but not to the extent Mr. Findley infers. The largest error present which could be attributed to these irregularities is 4 per cent and it occurs but twice in eighteen graphs.

The offset method of interpreting results from a load-deformation curve is a very desirable method. In its final analysis it is merely a refinement of the load-deformation test.

As stated in the preface, this paper deals with the methods of determination of allowable bearing pressure, not with the actual determination. For this reason it is of little consequence so far as this paper is concerned whether or not the allowable unit bearing pressure is calculated from the formula $S_b = P/1.57rl$ or $S_b = P/2rl$. Therefore it seems that while Mr. Findley's remarks along this line are of value in explaining the reasons for the calculation of an average stress rather than a specific stress in determining bearing strengths, they are not pertinent.

As regards Table 4, the author is not quite clear as to what Mr. Findley regards as "sufficiently conclusive data." The fact that the curves were averages of several tests and that the plotted points were not shown should not make impossible the use of the offset method in certain cases, for it is a method applicable to any curve. If it is applicable to one curve, witness Table 4, it should be applicable to practically any curve. As a matter of record the author was able to fill in several of the blanks in Mr. Findley's Table 4 by the application of the offset method.

It is not understood what Mr. Findley means by "consistent results" as regards the results given by the offset method at 0.2 per cent, inasmuch as there is only one curve to show the performance under loading of the specimens conditioned in each of the three ways. These curves can certainly not be considered to be consistent with themselves, and it would be entirely out of reason to expect the different type materials to react in a like manner when subjected to loading.

Applications and Unusual Physical Properties of Synthetic Rubbers

By OTIS D. COLE,[1] AKRON, OHIO

This paper deals with some of the applications and more or less unusual physical properties of several of the commercial rubber-like synthetic materials on the market or in advanced stages of development at the present time. The materials which are considered as "synthetic rubbers" fall into four classes: (*a*) Buna S types or copolymers of butadiene and styrene; (*b*) Buna N types, or copolymers of butadiene and acrylonitrile; (*c*) Neoprenes, or polychloroprenes, or copolymers of chloroprene with other materials; and (*d*) Butyl rubber, a copolymer of isobutylene and butadiene. The Thiokols or organic polysulphide polymers might be included among the "synthetic rubbers" by some, but, although they appear to be vulcanizable, their thermoplasticity makes them useful only to a limited degree in places where elasticity and other rubber-like properties at higher temperatures are desired. Details concerning Buna S type polymer, adopted by the Rubber Reserve Company for use in tires and as a general-purpose rubber, are discussed. Some interesting data on the efficiencies of various synthetic rubber compounds as compared to those of natural rubber compounds at high and low temperatures are shown. Data are also given which indicate that the proper choice of plasticizer is important for compounds to be used for oil-resistant gaskets, sealing rings, and the like.

WITH over 90 per cent of the sources of our crude-rubber supply in the hands of the Japanese, and the shipping facilities for the remainder more or less in jeopardy, the rubber situation in this country is really serious. Were it not for the foresight of the managements and the unremitting labor of the research and technical staffs of the various organizations concerned in the development of the so-called synthetic rubbers, the future would look much more dismal than it does at present.

Search for Rubber-Like Materials

In the last few years the activity in research laboratories all over the country has been increasing steadily, and several materials have been developed which are approximately equal to natural rubber in most respects and for some purposes are better than the natural product. These materials are the result of systematic testing and selection of literally thousands of experimental materials in each class until only a few which have the desired characteristics remain. Each of these was selected for commercial use because of certain outstanding characteristics which make it desirable. So far, the producers are to be commended for the relatively few materials which have been introduced to the industry. This fact is more readily appreciated when one considers that it is possible more or less to "tailor-make" a material for each special purpose. It is appreciated greatly by the rubber compounders who are in the throes of a revolutionary change in thinking from terms of natural to those of synthetic rubber.

There are numerous articles in the literature on the preparation and compounding of various types and varieties of synthetic rubbers and the characteristics of the products made from them. A comprehensive summary of the literature up to June, 1940, was given by Wood (1).[2] Other more recent contributions are those of Schade (2), Street and Ebert (3), Street and Dillon (4), and Sebrell and Dinsmore (5). Comprehensive discussions of the processes involved and the economics of the synthetic-rubber industry are given by Cramer (6) and Bridgwater (7).

It should be pointed out that the term "synthetic rubber" is actually a misnomer. In order for the name to be true, it would be necessary for the synthetic materials to be identical chemically with natural rubber, which is not the case. These products should more properly be called "synthetic rubber-like materials." However, because of common usage they will be referred to in this paper as synthetic rubbers.

The idea of synthetic rubber is not new. As long ago as before and during the first world war the Germans were making a material called "methyl rubber" by the polymerization of dimethylbutadiene. This material was far from satisfactory in its properties, but was rather closely related chemically to natural rubber, which can be considered a polymer of methylbutadiene, or isoprene. Because of the failure of such a closely related material, the work on synthetic rubber resolved itself into a search for materials which, regardless of chemical structure, had the flexibility, strength, and elasticity of natural rubber, with or without its shortcomings. This search has led to the discovery (in some cases accidental) and development of the class of materials we know today as synthetic rubbers.

Classes of Synthetic Rubber

In addition to the class of synthetic rubber-like materials, there are available other materials which have some rubber-like characteristics. In this group are found organic polysulphide polymers (Thiokol), Vinylite, plasticized polyvinyl chloride (Koroseal), polybutylene (Vistanex), etc. These materials do not have the resilience that is characteristic of rubber, and they are "thermoplastic," even when compounded with other materials, instead of being "thermosetting" as in the case of compounded rubber. The plasticized Vinylites have some properties such as chemical inertness which make them more desirable in some cases where flexibility without resilience is required. The Thiokols or organic polysulphide polymers might be included among the synthetic rubbers by some, but, although they appear to be vulcanizable, their thermoplasticity makes them useful only to a limited degree in service where elasticity and other rubber-like properties at elevated temperatures are desired. Thiokol is one of the oldest of the American types of synthetic materials having some rubber-like properties. Its high resistance to oils, other solvents, and all kinds of aging, as well as its impermeability to gases makes it serve a very useful purpose in industry in spite of its thermoplasticity.

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[1] Research Laboratory, The Firestone Tire & Rubber Company.

Contributed by the Rubber and Plastics Subdivision of the Process Industries Division and presented at the Semi-Annual Meeting, Cleveland, Ohio, June 8–10, 1942, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

The materials which can be considered as synthetic rubbers fall into four classes: Neoprene types, Buna S types, Buna N types, and Butyl rubber.

Neoprene Types. The Neoprene types are polymers of chloroprene or copolymers of chloroprene with other materials. It is evident that there is possible a wide variety of products all made from the same basic materials. The term "Neoprene" therefore should not be considered as descriptive of a single product. Several types of Neoprene have appeared on the market and undoubtedly others will appear as specialized service demands them. The Neoprenes as a class may be termed as moderately oil-resisting rubbers.

Buna S Types. The Buna S types are copolymers of butadiene with styrene. A Buna S type of polymer has been adopted by the Rubber Reserve Company for use in tires and as a general-purpose rubber. Fundamentally, it is less efficient and develops lower physical properties at ordinary temperatures than natural rubber. However, it is much more heat-resistant, and, as will be shown later, when properly compounded is practically as cold-resistant as natural rubber. It is similar to natural rubber in that it is not oil-resistant.

Buna N Types. The Buna N types are copolymers of butadiene and acrylonitrile. They are generally oil- and hydrocarbon-solvent-resistant to a high degree. However, they are poor in resistance to oxygenated solvents. Their cold resistance in general is poor but by correct compounding fair to good cold resistance can be obtained. A large number of different varieties of this type are on the market under the trade names of Hycar O. R., Chemigum I, II, III, and X, Perbunan, Buna[3] NM, NX, NXM, NXX, and NF, etc. These vary as to the ratios of the constituents and the method of preparation, but their properties as a class are similar.

Butyl Rubber. Butyl rubber is reported to be a copolymer of isobutylene with butadiene. It is high in chemical stability, it is very impermeable to gases, and is reported to be heat-resistant. It may eventually have the lowest cost of all the synthetic rubbers. It is poor in oil resistance and contrary to some recent claims in the press the present material is not satisfactory for tires. Because of the relatively small amount of butyl rubber which has been produced and made available for experimental purposes, this material has not been evaluated as completely as some of the other synthetic rubbers. In the near future a new plant will go into production and at that time sufficient material will be made available, so that a complete evaluation of all its properties in all types of products can be made.

It should be understood that with the rapid development now going on in this entire field, the possibility of a new material, which surpasses all the present ones now in production or development, being found is not remote but is very likely. The following examples of some of the more or less unusual properties of a few of the present materials are given with the foregoing in mind. New developments may greatly change the picture in the next few months.

When evaluating synthetic rubbers in the laboratory, it must be kept in mind that we are working with new, entirely different materials and not rubber. Laboratory evaluation tests for rubber are the result of years of experience in applying and comparing the results of such tests to performance of compounds in service. It is easy to make the mistake of attempting to interpret laboratory data on synthetic rubbers on the basis of experience with natural rubber, and thus drawing erroneous conclusions.

Because of our present interest in the butadiene or Buna type synthetics, the majority of the examples will be chosen from that class. The Buna synthetic rubbers chosen for examples were:

1 Buna S, a modified butadiene-styrene copolymer.
2 Buna NM, a modified butadiene-acrylonitrile copolymer.
3 Buna NXM, a modified butadiene-acrylonitrile copolymer.
4 Buna NF, a modified butadiene-acrylonitrile copolymer.

Each of these types has been developed to fit a special need, such as ease of processing, ease of molding, oil resistance, cold resistance, etc.

[3] The present trade names for the Buna N type synthetic rubbers mentioned in this paper specifically as Buna NM, Buna NXM, etc. are Butaprene NM, Butaprene NXM, etc.

Tensile Strength

If we were to have based our conclusions as to the quality of the Buna rubbers on results of tensile-strength tests, they would not have been considered further after the first tests. The data given in Table 1 show that the tensile strengths of pure gum stocks prepared from the Buna-type rubbers are very low. It is only after these rubbers have been compounded with carbon black that they exhibit tensile properties which remotely resemble those of natural rubber stocks. Even then the tensile strengths are far below those of natural rubber stocks having similar compounding. Neoprene is different in this respect, for the pure gum stock has higher tensile strength than the stock compounded with channel black. It is evident that tensile strength is not the complete criterion for the judgment of quality of Buna type synthetic rubbers, for if it were, none of them would have been considered further as possible materials for replacement of natural rubber in tires, for example.

TABLE 1 COMPARISON OF TENSILE DATA FOR VARIOUS RUBBERS WITH SIMILAR COMPOUNDING

Type of rubber	Tensile strength,[a] psi		
	Pure gum	P-33 black[b,c]	Channel black[c,d]
Natural	2500	...	4400
Buna S	200	1200	3000
Buna NM	700	1400	2500
Buna NXM	800	1600	2700
Buna NF	450	800	1300
Neoprene	3900	...	3400

Type of rubber	Stress at 400 per cent elongation, psi		
	Pure gum	P-33 black[b]	Channel black[d]
Natural	650	...	2100
Buna S	...	1100	1900
Buna NM	550	1025	1450
Buna NXM	550	1225	1750
Buna NF	...	825	...
Neoprene	950 (600%)	...	2200

Type of rubber	Elongation at break, per cent		
	Pure gum	P-33 black[b]	Channel black[d]
Natural	800	...	600
Buna S	300	430	580
Buna NM	420	520	560
Buna NXM	480	500	550
Buna NF	300	400	340
Neoprene	1020	...	550

[a] All stocks were cured to the "optimum cure" as judged by tensile strength.
[b] These stocks were compounded with softeners, vulcanizing ingredients etc., and contained 100 parts of P-33 black per 100 parts of rubber or synthetic rubber.
[c] The tensile strengths of the various Buna N type stocks containing P-33 and channel black can be measurably increased, or decreased, by the use of different softeners.
[d] These stocks contained softeners, vulcanizing ingredients, and 50 parts of rubber grade channel black per 100 parts of rubber or synthetic rubber.

The tensile strengths of the various Buna N type synthetic rubbers show rather definite correlation with other physical properties, such as oil resistance, cold resistance, etc., as will be shown later.

It should be pointed out that by special handling during mixing, the tensile properties of the Buna types can be improved, but experience has shown that the improvement in most service characteristics is generally so small that it does not justify the extra time, trouble, and expense.

Oil Resistance

The fact that the Buna N types of synthetic rubber exhibit very low swelling when exposed to oils is one of the most unusual and at the same time the most important property of this group. As a class they are better than Neoprene in this respect. The different Buna N types have different oil resistances as is shown in Table 2. (For the sake of comparison, figures for Neoprene GN are included in this table.)

TABLE 2 PERCENTAGE OF INCREASE IN VOLUME[a] AFTER 24 HOURS IN DIFFERENT OILS

Type of synthetic rubber	Paraffin-base S.A.E. 10 oil	Kerosene 80 per cent, benzene 20 per cent
Buna S	152	..
Buna NM	3	16
Buna NXM	—3	12
Buna NF	40	42
Neoprene GN	23	91

[a] Volume change determined by the method of Garvey (8).

The data show that the resistance to swelling in oil may be controlled, more or less, by proper selection of Buna N type synthetic rubbers. The negative swelling, or shrinking, of the Buna NXM stock in S.A.E. 10 oil is most probably due to the extraction of some of the plasticizer by the oil. This property is important and must be compensated for by proper compounding when shrinkage is not desirable. By correct compounding, it is possible to produce a stock which will not change in volume when immersed in oil. This is done by selection of Buna N types and plasticizers.

An interesting and rather unusual swelling phenomenon is illustrated by the following example:

Two Buna NXM gasket compounds *A* and *B* were tested for resistance to swelling in carbon tetrachloride, which has a very pronounced swelling effect on all of the Buna N types. Compound *A* contained, as the plasticizer, an oil which was soluble, while compound *B* contained a resin which was practically insoluble in carbon tetrachloride. The data for percentage of swelling versus time of immersion for the two stocks are shown in Fig. 1. It is obvious that compound *A* would not be satisfactory as a gasket in contact with carbon tetrachloride. These two compounds in carbon tetrachloride are given as examples because the effect is exaggerated. However, our observations would indicate that this phenomenon is rather common, especially with aromatic solvents, and may be the cause of actual shrinkage in other oils. It is interesting to note that the usual 3-day or 7-day immersion test would have rated these two compounds as equal in resistance to swelling.

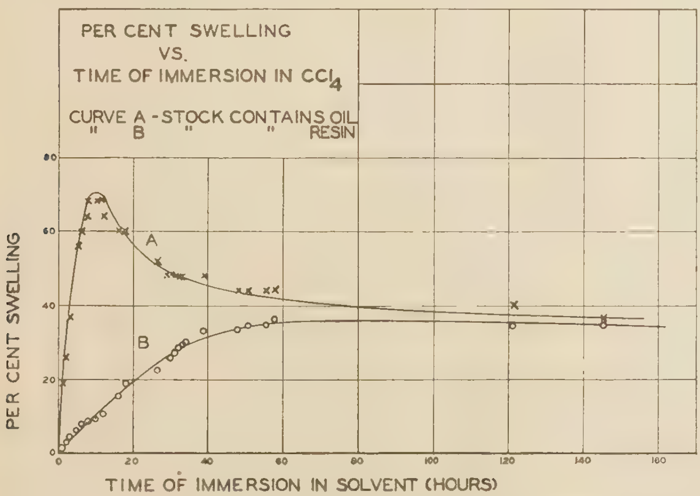

FIG. 1 PERCENTAGE OF SWELLING VERSUS TIME OF IMMERSION IN CARBON TETRACHLORIDE
(Curve *A*, stock contains oil; curve *B*, stock contains resin.)

The shape of the curve for stock *A* may be explained by the assumption that during the first few hours the solvent is imbibed in the compound and goes into solution in the oil plasticizer, thus causing it to swell rapidly. The solution of solvent in oil is then extracted and at equilibrium only the effect of the solvent on the Buna NXM compound remains. Since the plasticizer and swelling solvent are mutually insoluble, the shape of the curve for compound *B* is the normal shape for a Buna NXM compound with no plasticizer. It is suggested that it might be well in some cases to observe the actual course of swelling rather than merely the final results.

The method of Garvey (8) lends itself more readily to following changes in swelling during immersion than does the regular A.S.T.M. method (9).

Resilience or Rebound Efficiency

The property of resilience is one which makes rubber an unusual material. In general the Buna, or butadiene, types of synthetic rubbers normally have lower resilience than does natural rubber. Among the Buna N types, the rebound resilience at room temperature depends upon the specific type used. This is shown in Fig. 2, which shows rebound resilience data for pure gum compounds at various temperatures. The Buna S type illustrated in Fig. 2, shows a rebound value intermediate between Buna NM and NF at room temperature. The values for the three synthetic rubber pure gum compounds approach each other at high temperatures, but do not reach those of the natural rubber pure gum stock.

These rebound resilience values were obtained by the use of a falling-ball apparatus and technique developed by I. B. Prettyman of the Physics Division of the Firestone research laboratories.

Although the data shown in Fig. 2 would indicate that Buna S type of synthetic rubber is normally less efficient than natural rubber, it does not follow that it cannot be compounded to produce highly efficient stocks. However, when a stock is compounded to produce equal efficiency, the hardness is increased over that of a natural rubber compound. Fig. 3 shows a comparison between a high-efficiency rubber compound and a Buna S compound, designed to have approximately the same efficiency. The hardness values at the various temperatures are included to illustrate the foregoing point.

A comparison of the rebound efficiencies of typical natural rubber, Neoprene GN and Buna S high channel black compounds is shown in Fig. 4. These curves corroborate other efficiency tests which show that Buna S has a higher hysteresis loss than do natural rubber and Neoprene GN. Natural rubber and Buna S in these stocks are practically identical at low temperatures, but show a wide variation at room temperatures and above. The curve for the Neoprene GN stock is practically parallel with that for natural rubber through the entire temperature range.

Cold Resistance

The property of cold resistance, or the lack of it, in the oil-resistant synthetic rubbers is of utmost importance to those who

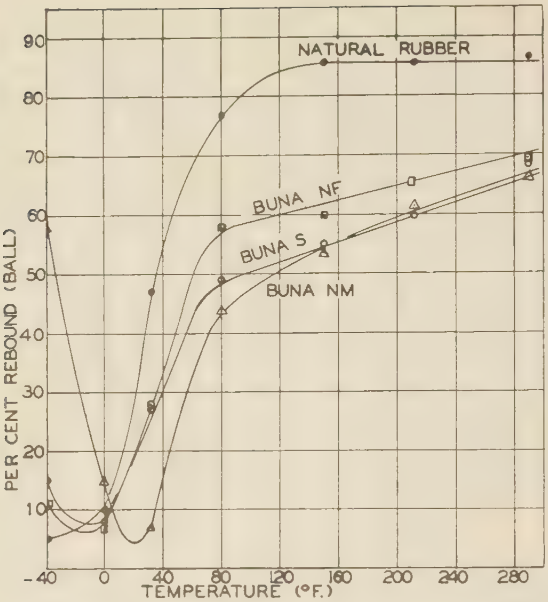

FIG. 2 EFFICIENCIES OF VARIOUS BUNA TYPES AND NATURAL RUBBER PURE GUM STOCKS AT DIFFERENT TEMPERATURES

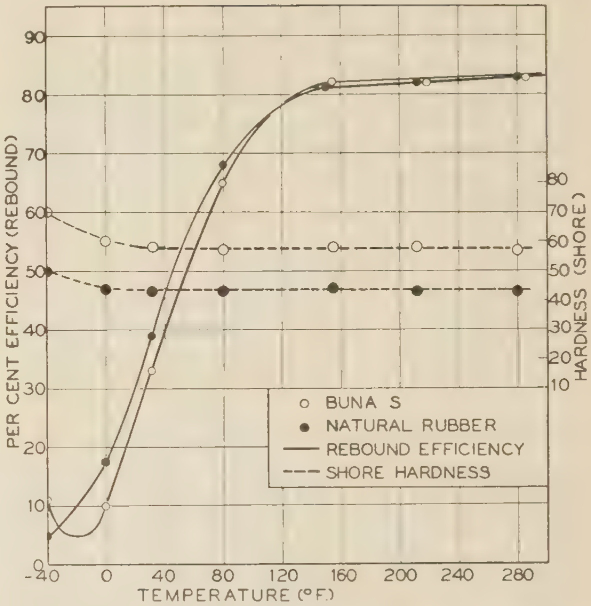

FIG. 3 EFFICIENCY AND HARDNESS OF RUBBER AND SPECIALLY COMPOUNDED BUNA S STOCKS AT VARIOUS TEMPERATURES

are designing or specifying parts where these rubbers are to be used. The increase in resilience at very low temperatures may be caused by changes in hardness as is shown in Fig. 3. So far, no definite relationship between the shape of the resilience-temperature curve and the actual "brittle point" has been observed. It should be pointed out that the minimum point in the resilience-temperature curve does not mean that, at that temperature, the rubber is brittle and will break upon bending. There are some indications (see Fig. 3) that these minima mark the beginning of the definite stiffening or hardening which eventually renders the material unserviceable at lower temperatures.

In general the cold resistance of the highly oil-resistant rubbers is poor, the more oil-resistant the rubber, the poorer the cold resistance. The data given in Table 3 show the temperatures at which several Buna type and rubber compounds, varying only in the rubber used, become frozen or stiff to the touch. Refrigerated methyl alcohol was used as the coolant in this case. Values for the brittle point and "freezing point" taken from the literature (10, 11) are given for comparison.

The figures given in Table 3 differ from those given in the

TABLE 3 STIFFENING, FREEZING, AND BRITTLE POINTS FOR NATURAL AND SYNTHETIC RUBBERS

Type of rubber	Stiffening point, deg F	Brittle[a] point, deg F	Freezing[b] point, deg F
Natural	—48 to —52	—69.7	—86.8
Buna S	—36 to —40	—86.8 to —94	—86.8[c]
Buna NF	—40	......	...
Buna NM	—9	—51.7[c]	...
Buna NXM	.....	—14.8 to —18.4[c]	...
Neoprene GN	—33	......	...

[a] Refer to Bib. (10).
[b] Refer to Bib. (11).
[c] These stocks were undoubtedly prepared from different Buna type synthetic rubbers from those reported under "stiffening point."

literature in that they are higher in every case. However, the present figures are based on the observer's judgment as to when the compounds became too stiff to be serviceable, while those of Selker, Winspear, and Kemp were the temperatures at which the compounds actually could be broken because of brittleness, and those given by Koch were the temperatures at which the compounds just began to soften after being frozen solid. All comparisons of data should be made within one column of the tabulation and not between different columns.

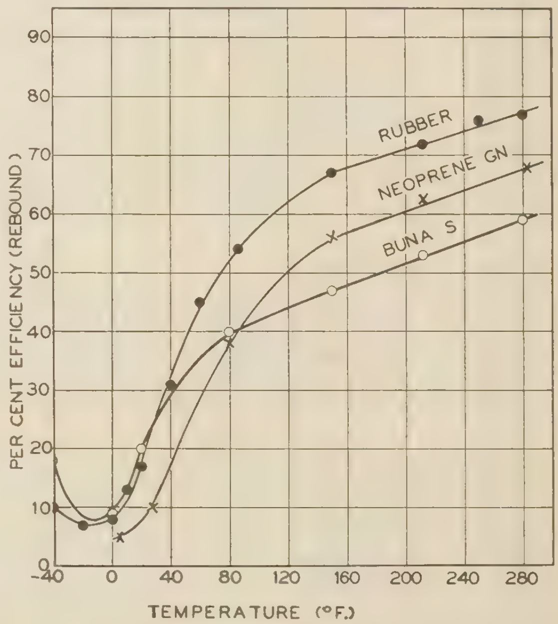

FIG. 4 EFFICIENCES OF RUBBER, NEOPRENE GN, AND BUNA S COMPOUNDS CONTAINING HIGH CHANNEL BLACK AT VARIOUS TEMPERATURES

It is well to mention here that in certain cases where rubber parts are to be used in contact with a solvent in service, the cold-resistance tests should be made using the same solvent, refrigerated, as the coolant. For example a fuel-pump-diaphragm compound when tested in air was frozen solid at —15 F but when immersed in the fuel it was perfectly flexible at —50 F because of the plasticizing action of the fuel.

The use of plasticizers plays an important part in the attain-

ment of good cold resistance with the Buna N and Neoprene types of synthetic rubbers. Fig. 5 shows the effect of various mixtures of two plasticizers *A* and *B* on the temperature at which several Buna NM and Buna NXM compounds ceased to become serviceable in air. It will be noted that Buna NM is more cold-resistant in all cases than Buna NXM with the same plasticizer mixture present. It is possible to prepare compounds from Buna NF, treated with the proper plasticizer, which will be serviceable at temperatures as low as —70 F.

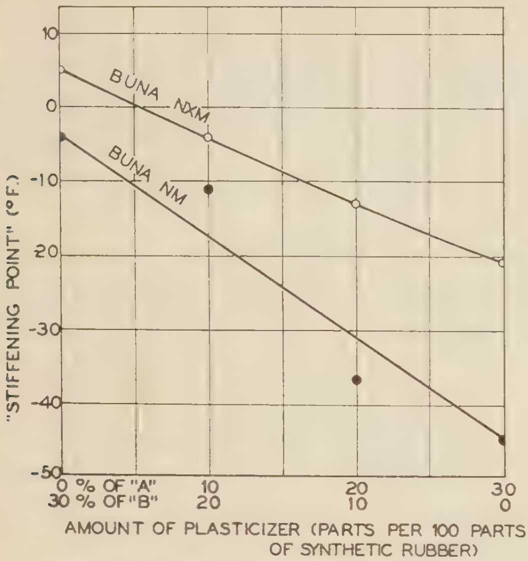

Fig. 5 Effect of Various Mixtures of Plasticizers *A* and *B* Upon the "Stiffening Points" of Different Buna N Type Synthetic Rubber Compounds

Resistance to Heat and Aging

An outstanding and rather unusual property of most of the synthetic rubbers, especially those of the Buna and Neoprene types, is their remarkable resistance to decomposition by heat. An apparatus designed by the physics division of the Firestone research laboratories was used to determine the decomposition temperatures of several natural-rubber, Buna, and Neoprene type stocks under carefully controlled conditions. A discussion and more thorough description of the apparatus and method is published elsewhere.[4] These thermal decomposition temperatures are shown in Table 4. The remarkably high heat resistance of Buna S as compared to natural rubber and the Buna N types would suggest its use for high-temperature service where oil resistance is not a serious factor. Neoprene GN also shows superior heat resistance. However, all of the Buna and Neoprene type synthetic rubbers become harder during continued exposure to high temperatures but it is possible in some cases to plasticize them to overcome this. This will tend to lower the heat resistance slightly.

TABLE 4 THERMAL DECOMPOSITION TEMPERATURES FOR NATURAL AND SYNTHETIC RUBBER COMPOUNDS

Type of rubber	Thermal-decomposition temperature,[a] deg C
Natural	194 – 198
Buna NF	220 – 229
Buna NM	220 – 229
Buna NXM	205 – 212
Buna S	Above 249
Neoprene GN	239 – 249

[a] Temperature at which a rubber compound will decompose in 1 hr.

[4] This description was published in a paper, "Thermal Decomposition of Natural and Synthetic Rubber Stocks," by I. B. Prettyman and presented at a meeting at Buffalo, N. Y., Sept. 11, 1942, of the Rubber Division of the American Chemical Society.

The synthetic rubbers show exceedingly good resistance to all types of accelerated aging. A comparison of the accelerated-aging characteristics of the various Buna types illustrated in Table 1 is given in Table 5. Their resistance to heat (air bomb) is remarkable when it is remembered that similar natural rubber compounds are badly deteriorated after only 10 hours under the same conditions. As the data for oxygen-bomb aging indicate, the Buna N types show slightly less resistance to oxidation than to other types of aging. However, their relative deterioration is much less than that of rubber under the same conditions. Buna S shows some susceptibility to oxidation, but is affected much less than natural rubber.

TABLE 5 PERCENTAGE OF ORIGINAL TENSILE STRENGTH AND ELONGATION RETAINED BY DIFFERENT BUNA TYPE SYNTHETIC RUBBER COMPOUNDS[a] AFTER VARIOUS TYPES OF AGING

	Type of aging					
	Air bomb[b]		Oxygen bomb[c]		Oven[d]	
Type of synthetic rubber	Tensile	Elong	Tensile	Elong	Tensile	Elong
Buna S	105	77	100	98	94	60
Buna NM	114	65	93	94	116	63
Buna NXM	112	64	103	94	118	68
Buna NF	151	70	106	83	150	73
Natural rubber (for comparison)	...	..	56	..	91[e]	..

[a] The original values for these compounds are given under "P-33 black" in Table 1.
[b] Aged 15 hr in air bomb at 260 F; 80 psi air pressure.
[c] Aged 46 hr in oxygen bomb at 158 F; 300 psi pressure.
[d] Aged 28 days in hot-air oven at 158 F.
[e] After only 14 days in oven.

All the synthetic rubbers harden or become stiffer during heat-aging treatments as is illustrated by the elongation values in Table 5. This fact must be considered when designs are being made for the use of these materials. All natural-aging data which have been obtained up to this time would indicate that the Buna type synthetic rubbers will be entirely satisfactory in this respect.

Although no data for Neoprene are given here, its high resistance to aging is well known. However, it stiffens during heating and aging to an equal if not greater degree than the Buna types.

Use of Synthetic Rubbers

The Buna S types can be used in nearly all cases to replace natural rubber. Generally satisfactory service as compared to natural rubber has been observed in the case of passenger-car tires. However, it must be pointed out that, with this or any synthetic rubber, as in the case of any new material, certain difficulties are present and must be overcome. For example, truck and heavy-duty tires at present are the result of years of specialized engineering and design, based on the characteristics of natural rubber. In order to meet the service conditions encountered today, it may be that some radical departure from present thinking and designs may have to be made in order to fit synthetic rubber into the picture. It is sufficient to say that the Buna S types appear to fit more nearly into the scheme of things as we see them today than any of the other synthetic rubbers.

The Buna N types of synthetic rubbers are being recommended for all types of mechanical goods such as hose, packings, gaskets, rollers, etc., wherever high resistance to aging, oils and hydrocarbon solvents is desired.

The Neoprene types are used wherever moderate oil resistance and good resistance to ozone and sunlight are desired. Since it will not support combustion it is widely used as a flameproof, oilproof, wire insulation.

Butyl rubber can best be utilized where its inherent properties

of chemical stability, resistance to ozone, light, and high impermeability to gases can be used to best advantage. As has been pointed out no entirely satisfactory tires have as yet been made from Butyl rubber.

One of the major problems which is not stressed in the discussions of synthetic rubber in the press is that of processing. It has been said that synthetic rubbers process just like natural rubber on regular rubber machinery. On the contrary, however, each material behaves differently and, although the difficulties are not insurmountable, even the training of personnel to handle each new material involves considerable time and effort.

Neoprene GN and some of the Buna N types have been found to give results in tire tread wear tests equal to, or better than, those given by natural rubber. However, several major difficulties in processing and fabrication have caused the trend toward a Buna S type for tire rubber.

Many of the really serious problems of today may be problems only because of the fact that certain handling processes designed for natural rubber are not fundamentally correct for the synthetic rubbers used. Because of this, it is possible that in the future the rubber industry will depend more than ever upon the mechanical engineer for the solution of its processing and handling problems.

Acknowledgment

The writer wishes to acknowledge the helpful advice of Dr. J. N. Street and the assistance of various members of the Research Department of the Firestone Tire and Rubber Company, in collecting and assembling the material for this paper, especially F. T. Donaldson and J. R. Rafter. He wishes to thank the Firestone Tire & Rubber Company for permission to present this paper.

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Metal Cutting With Abrasive Wheels

By W. B. HEINZ,[1] BOUND BROOK, N. J.

The analytical investigation herewith reported was undertaken in order to determine whether or not the manner in which an abrasive cutoff wheel is operated might have an important effect upon the amount of service obtainable during the life of each wheel; and whether or not the maximum possible cutting rate might also be subject to improvement by better methods of operation. The geometrical relationships involved in cutting round stock have been determined, with the stock rigidly clamped and with the stock rotating in each direction relative to the wheel rotation. Based on the geometry of the several conditions, some predictions have been made as to the advantages which might be gained by a well-organized experimental program.

Nomenclature

The following nomenclature is used in this paper:

A = cross-sectional area of cut
ΔA = number of square inches which have been cut by arc of wheel from 0 to ϕ during time required for a point on wheel to travel through angle ϕ.
ΔA_b = same as ΔA, when work is rotating in backward[2] direction
ΔA_f = same as ΔA, when work is rotating in forward[2] direction
ΔA_p = "packing area:" maximum theoretical area of cut which can be accomplished by one pass of wheel face length $R\,\phi$
ΔA_{pb} = same as ΔA_p, when work is rotating in backward[2] direction
ΔA_{pf} = same as ΔA_p, when work is rotating in forward[2] direction
$\dot{A}$ = dA/dt
C = $R + r_2$
F = symbol meaning "a function of"
N = $C\,\omega/\Omega$
R = radius of the abrasive wheel; becomes smaller as wheel wears
r = radius from center of work rotation to any point in arc of contact between work and wheel (see Fig. 1)
r_1 = radius of round stock to be cut
r_2 = radius from center of work rotation to bottom of cut at the point just leaving contact with wheel (see Fig. 1)
t = time
Δx = depth of cut made by arc of wheel from 0 to ϕ during time required for a point on wheel to travel through angle ϕ
x_{db} = "drag distance:" average distance an increment of metal is dragged, from point where it was cut loose to its exit from cut, when work is rotating in backward direction
x_{df} = same as x_{db}, when work is rotating in forward direction
x_{d0} = same as x_{db}, when work is stationary
$x_{db\,\text{avg}}$ = "average drag distance:" average of all instantaneous values of drag distance x_{db} (backward), during a complete cut through work
$x_{df\,\text{avg}}$ = same as $x_{db\,\text{avg}}$ except for forward work rotation
$x_{d0\,\text{avg}}$ = same as $x_{db\,\text{avg}}$ except when work is stationary
y_b = amount by which cut at a single point (at $R\phi$) is deepened when wheel turns through an increment of angle $\Delta\,\phi$ in backward direction
y_f = same as y_b, with forward work rotation (see Equation [7])
α = angle of rotation of work about its own axis (radius). See Fig. 1
ϕ = angle about wheel axis, Fig. 1
ϕ_m = maximum value of angle ϕ at any instant
ϕ_p = angle at which wheel surface voids would theoretically become completely filled with removed metal
ϕ_{pb} = same as ϕ_p, for backward work rotation
ϕ_{pf} = same as ϕ_p for forward work rotation
θ = angle about work axis, Fig. 1
θ_m = maximum value of angle θ at any instant
Ω = angular velocity of wheel ($d\phi/dt$) in radians per second
ω = angular velocity of the work ($d\alpha/dt$) in radians per second.
σ = wheel "porosity," measured in inches. The theoretical maximum depth of metal which any part of the wheel can remove at one pass. $\sigma = \dot{A}_{\max}/R\Omega$

Introduction

The object of the work described in this paper is to predict how an abrasive wheel should be operated so that it will provide the best combination of production rate and total production before it wears out. Particular attention has been directed to the cutting of large-sized bars, since the rate of wheel wear on most cutoff machines is greater when cutting stock of large size.

An abrasive wheel is commonly visualized as presenting a large number of minute cutting edges which are kept sharpened by the continual breaking off of dulled points. As each point cuts out chips of metal, these chips must accumulate in the clearance spaces adjacent to the point until they have been dragged to the limit of the cut, where they are thrown from the wheel.

The breakage of points and consequent wheel wear may occur partly from overstressing while actually cutting, but probably most of such breakage results from combined bending and impact against loose chips which accumulate toward the end of the cut. These chips undoubtedly do not stay neatly confined within the surface voids of the wheel, but some of them are rolled and dragged along beneath the wheel points until they can finally escape. The destructive influence of such random particles is believed to be the most important cause of rapid wheel wear. Consequently, the longer the average distance through which the loosened metal must be dragged before it is thrown out of a cut, the shorter will be the life of the wheel.

How much can this average "drag distance" be reduced by keeping the work moving instead of leaving it stationary as the wheel advances? Answers to this question can be predicted by

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[1] Consulting Engineer. Mem. A.S.M.E.

[2] With "backward" work rotation, adjacent points on wheel and work are moving in the same direction; and vice versa.

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an analytical study of the cutting action of the wheel on fixed work and on work which rotates. In each case the axes of the wheel and the work are assumed to be approaching each other radially.

Analysis of Rotated Work

The geometry of a wheel cutting rotated work is given in the diagram, Fig. 1. A wheel of radius R is cutting into round stock of radius r_1 at a rate (square inches per second) which is assumed to be constant. The wheel has cut into the work to a radius r_2 at

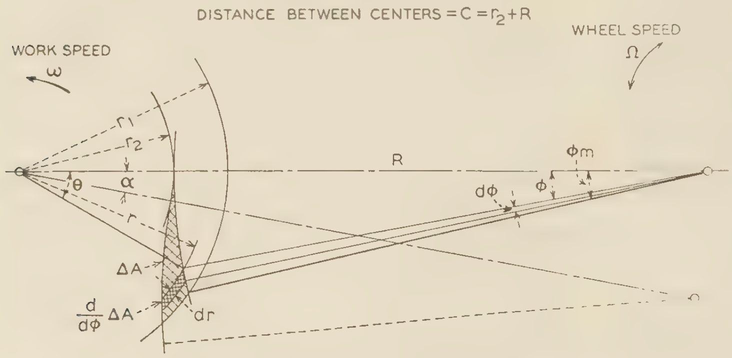

Fig. 1 Geometrical Diagram of Abrasive Wheel Cutting a Rotating Cylindrical Bar

which it will cut steadily, without progressing farther toward the center, except as the rotation of the work allows it to spiral inward. Work and wheel radii intersecting at any radius r in the work intercept the angles θ and ϕ, respectively.

For geometrical simplicity, the angle α is shown as representing the advance of the wheel as its axis rotates about the work axis. This is equivalent to rotation of the work about its own axis, but more readily illustrated. The cross-hatched section represents the metal which has been removed, as the wheel advanced through the angle α. The wheel and the work are rotating at constant speeds Ω and ω (radians per sec), respectively.

It is evident from Fig. 1 that

$$\text{Length of cut} = R\,\phi \qquad [1]$$

$$\text{Depth of cut} = (r - r_2) = R(1 - \cos\phi) + r\,(1 - \cos\theta) \qquad [2]$$

The area of the annulus between the radii r_2 and r is $\pi\,(r^2 - r_2^2)$. This area would be cut in the time $2\pi/\omega$ sec. Consequently, the instantaneous rate of cut is

$$dA/dt = \dot{A} = (\omega/2)\,(r^2 - r_2^2) \qquad [3]$$

in square inches per second.

From the foregoing equations, Equation [4] is derived

$$\Delta A = RC\,\alpha\,(1 - \cos\phi) \qquad [4]$$

in which

$$C = R + r_2$$

The increment of area cut across the entire contact arc is given by Equation [4], where $\phi = \phi_m$. The instantaneous cutting rate is then

$$\dot{A} = RC\,\omega\,(1 - \cos\phi_m) \qquad [4a]$$

Since the work is rotating continuously as well as the wheel, while a point on the wheel is moving through the angle ϕ the angle α is also increasing. Consequently, the two angles ϕ and α are interrelated, as follows

$$\alpha = \omega\phi/\Omega \qquad [5]$$

Combining Equations [4] and [5], we obtain Equation [6] which tells us the number of square inches that have been cut by the arc of the wheel from 0 to ϕ, during the time required for a point on the wheel to travel through the angle ϕ

$$\left.\begin{aligned}\Delta A_f &= RC\,\phi\,(\omega/\Omega)\,(1 - \cos\phi)\\ &= RC\,\omega\phi^3/2\Omega \text{ approx}^3\end{aligned}\right\} \qquad [6]$$

Equation [6] only holds for directions of rotation such that adjacent points on wheel and work are moving in opposite directions, hereinafter termed "forward rotation." Backward rotation is defined as that condition when adjacent points on wheel and work are moving in the same direction.

If we were to plot Equation [6] on rectangular co-ordinate paper, it would appear as in Fig. 2. The equivalent cut depth at any angle would be the derivative of the area with respect to the distance $R\phi$ along the arc of contact

$$y_f = (d/dR\phi)\,\Delta A_f = 3C\omega\phi^2/2\Omega \qquad [7]$$

Now, if we move the axis toward the left to coincide with the end of the cut $-\phi_m$, the equation about this new vertical axis is as follows

$$y_b = \frac{3C\omega}{2\Omega}\,(\phi - \phi_m)^2 \qquad [8]$$

To determine the amount of area which is cut by the wheel arc subtending the angle ϕ from the new axis, corresponding to backward rotation, we must integrate Equation [8], obtaining

$$\Delta A_b = \int_0^{R\phi} y_b dR\phi = \frac{RC\omega}{2\Omega}\,(\phi^3 - 3\phi_m\,\phi^2 + 3\phi_m^2\phi) \qquad [9]$$

The wheel has only a certain amount of void space opening

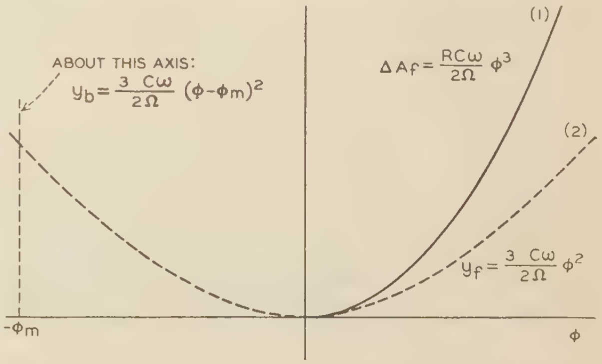

Fig. 2 Curves Illustrating Equations [6] and [7]

(1, Area cut by arc of wheel from 0 to ϕ while wheel is moving through angle ϕ in forward direction. 2, Amount by which cut at a single point [at $R\phi$] is deepened when wheel turns through an increment of angle $\Delta\phi$.)

into its cutting surface. As soon as that available space in any area of the wheel surface becomes completely filled with chips, that spot can no longer do any cutting. In practice the total space probably never does become completely filled; so that a measure of the voids should be regarded as a theoretical limit to the amount of material which any part of the wheel can remove during one pass. Suppose we call this limit the "porosity" of the wheel, represented by the symbol σ.

Then the maximum theoretical area of cut which can be accomplished by one pass of the wheel face length $R\phi$ is

[3] Obtained by substituting for $\cos\phi$ the first two terms of its infinite series, $1 - (\phi^2/2)$.

$$\Delta A_p = \sigma R \phi_p \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

Equation [11] is adapted from Equation [6] to give another expression for the "packing area" in terms of the "packing angle" ϕ_p at which the wheel would theoretically become packed with forward rotation

$$\Delta A_{pf} = \frac{RC\omega}{\Omega} \phi_{pf} (1 - \cos \phi_{pf}) \ldots\ldots\ldots\ldots [11]$$

On combining the two equations we have

$$\sigma R \phi_{pf} = \frac{RC\omega}{\Omega} \phi_{pf} (1 - \cos \phi_{pf})$$

$$\cos \phi_{pf} = 1 - \frac{\sigma\Omega}{C\omega} \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

Since the cosine of an angle is approximated closely enough for our purposes by the first two terms of its equivalent series, we can write

$$1 - \frac{\phi_{pf}^2}{2} = 1 - \frac{\sigma\Omega}{C\omega}$$

$$\phi_{pf} = \sqrt{\frac{2\sigma\Omega}{C\omega}} \ldots\ldots\ldots\ldots\ldots\ldots [13]$$

Equation [13], expressed in radians, represents the angle through which a spot on the wheel must theoretically travel in the "forward" direction in order to accumulate just enough metal to fill its voids completely.

A corresponding expression for the packing angle with backward rotation is derived from Equation [9]

$$\phi_{pb} = \frac{3}{2} \phi_m \pm \frac{1}{2} \sqrt{\frac{8\sigma\Omega}{RC\omega} - 3 \phi_m^2} \ldots\ldots\ldots [14]$$

To illustrate the physical significance of the foregoing equations a typical example will be worked out numerically.

Wheel radius R = 12 in.
Work radius r = 3 in.
Wheel speed Ω = 220 radians per sec (2100 rpm)
Work speed ω = 0.105 radian per sec (1 rpm)

In Fig. 3, the curves for forward rotation have been computed from Equation [6]. They represent the total area of cut which is made in the arc ϕ while a point on the wheel edge is traveling from 0 to ϕ, for each of several constant cutting rates $\dot{A}$.

The corresponding curves for backward rotation could have been computed from Equation [9], but it was easier to construct them graphically from the "forward" curves. This is done by plotting, downward from the maximum value at ϕ_m (working from ϕ_m toward the left), the ordinates to the corresponding forward curve at positions measured from 0 toward the right. The resulting "backward" curve is the same as its forward curve turned right to left and upside done.

The total area of cut ΔA, plotted in Fig. 3, was removed by $R\phi$ inches of wheel periphery. How much metal is accumulated by a unit length (1 in.) of wheel as it progresses from 0 to ϕ?

To simplify our reasoning, we consider each inch of wheel as if it acts the same as one of a series of single cutting blades, as on a milling cutter. Evidently each blade cuts out the same amount of metal as every other blade which passes through the angle ϕ during the time interval ϕ/Ω.

Consequently, we can write

(Cut per unit length of wheel, on traveling from 0 to $R\phi$) = $1/R\phi$ × (total cut between 0 and ϕ during time ϕ/Ω) = $\Delta A/R\phi$

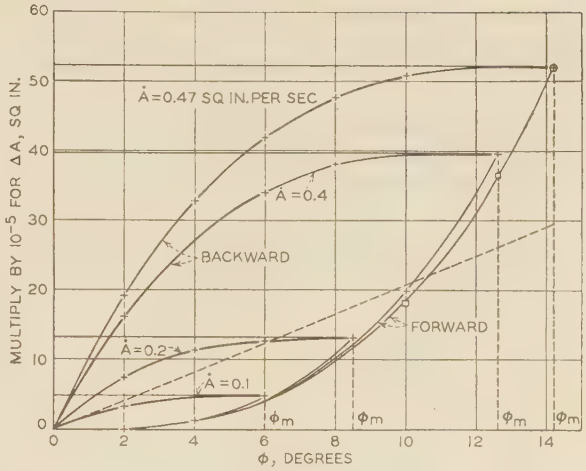

Fig. 3 Computed Curves Showing Number of Square Inches Cut by Arc of Wheel From 0 to ϕ While Wheel Is Moving Through Angle ϕ, at Various Constant Cutting Rates $\dot{A}$

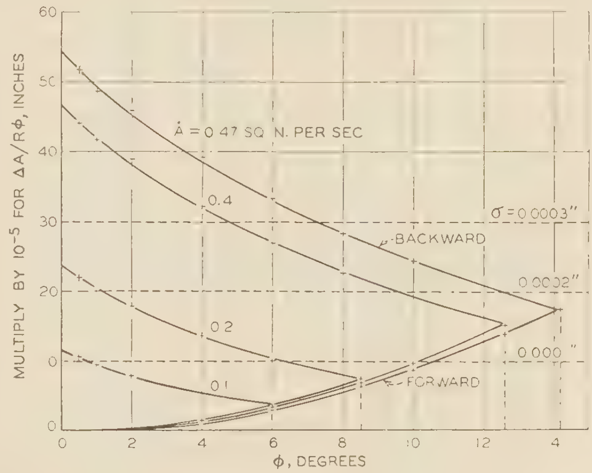

Fig. 4 Computed Curves Showing Average Area Cut by Each Peripheral Inch of Wheel as It Progresses From 0 to ϕ, at Various Constant Cutting Rates $\dot{A}$

Term ΔA is plotted in Fig. 3. Hence we divide each ordinate in Fig. 3, by the corresponding value of $R\phi$, and plot the quotients to obtain Fig. 4.

On comparing the curves of Fig. 4, with various wheel porosities σ, we interpret them as indicating the theoretical maximum cutting rates which are possible for wheels of stated porosities. Cutting is impossible in the region above the horizontal dotted line representing the wheel porosity.

For example, the theoretical maximum cutting rate $\dot{A}$, for a wheel of porosity 0.0002 in. (other conditions being as previously stated) is 0.17 sq in. per sec with backward rotation, and greater than 0.47 sq in. per sec with forward rotation. With nonrotating work, the theoretical maximum rate is the same as for forward rotation.

Conversely, we can compute the porosity of a wheel by determining its maximum cutting rate under specific conditions.

Wheel Service Factor

It seems reasonable to suppose that the rate of wheel wear depends on the following three factors, as well as on many others, which are well known:

1 Unit contact force.
2 Average carry of metal, or drag distance x_d.
3 Porosity of wheel, σ.

The cutting rate also depends on the same three factors, but inversely as the drag distance. Consequently the area that is cut while 1 sq in. of wheel is wearing away can be represented by the following ratio, some function of the foregoing factors

$$\frac{\text{Cutting rate}}{\text{Wearing rate}} = F\left[\frac{\dfrac{(\text{Unit force})}{(\text{Drag distance})}\cdot\sigma}{(\text{Unit force})\quad(\text{Drag distance})\times\dfrac{1}{\sigma}}\right]$$

$$\text{Service factor} = F\left(\frac{\sigma}{x_d}\right)^2 \ldots\ldots\ldots\ldots\ldots [15]$$

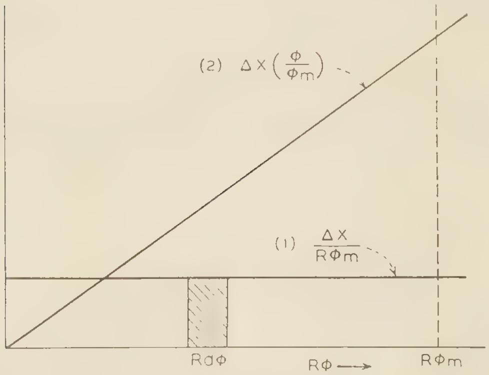

Fig. 5 Curves Illustrating Equations [16] and [17]

(1, Instantaneous depth of cut made by each peripheral inch of wheel as it progresses through fixed work. 2, Total cut made by each peripheral inch on moving from 0 to ϕ through fixed work.)

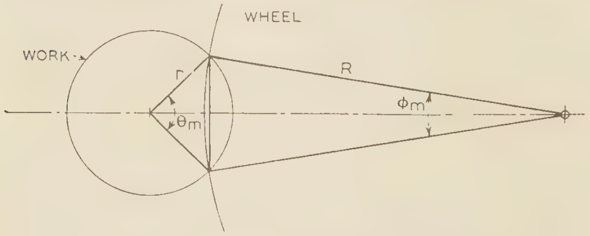

Fig. 6 Geometrical Diagram of Wheel Cutting Fixed Cylindrical Work

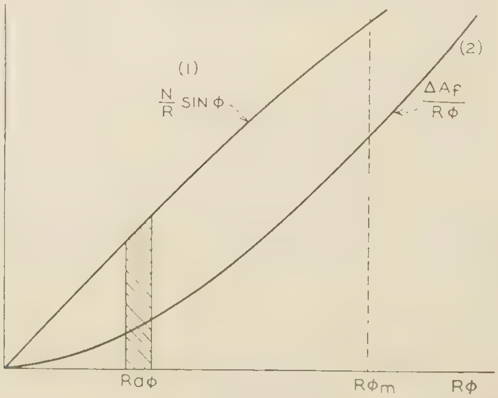

Fig. 7 Curves Illustrating Work Rotating Forward

(1, Instantaneous depth of cut made by each peripheral inch of wheel as it progresses through forward rotating work. 2, Total cut made by each peripheral inch of wheel moving from 0 to ϕ through forward rotating work.)

It appears that the wheel service factor (the ratio just given) is probably inversely proportional to the drag distance raised to some power greater than 1. What is the average drag distance when cutting off round stock?

Fixed Work. First, we must find the average distance the metal is carried in a cut of arc length ϕ.

The depth of cut made by each inch of wheel circumference ("wheel-inch") as it progresses (the "instantaneous cut depth") is $\Delta x/R\phi_m$.

This is the same as the area which is cut by each wheel inch when it moves 1 in., $\Delta A/(R\phi_m)^2$. We can equate the two

$$\Delta x/R\,\phi_m = \Delta A/(R\phi_m)^2 \ldots\ldots\ldots\ldots\ldots [16]$$

Also, the total cut per wheel inch on moving through the angle ϕ is

$$(R\phi)\,(\Delta x/R\phi_m) = \Delta x(\phi/\phi_m) \ldots\ldots\ldots\ldots [17]$$

Equations [16] and [17] are illustrated in Fig. 5.

The average distance the metal is carried is

$$x_{d0} = \frac{1}{\Delta x}\int_0^{\phi m} R(\phi_m - \phi)\left(\frac{\Delta x}{R\phi_m}\right) Rd\phi$$

$$x_{d0} = R\phi_m/2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots [18]$$

We must then determine the average cutting arc, as the wheel progresses through the bar. Fig. 6 illustrates the situation. The angles are related as follows:

$$R\sin(\phi_m/2) = r\sin(\theta_m/2) \ldots\ldots\ldots\ldots [19]$$

The average of $R\sin(\phi_m/2)$ as ϕ_m varies is the same as the average of $r\sin(\theta_m/2)$ from $\theta_m = 0$ to $\theta_m = \pi$.

$$R\sin(\phi_m/2)_{\text{avg}} = \frac{r}{\pi}\int_{\theta m=0}^{\theta m=\pi}\sin(\theta_m/2)\,d\theta_m$$

$$= \frac{2r}{\pi}\Big(-\cos(\theta_m/2)\Big)_0^{\pi} = 2r/\pi$$

Since $\sin\phi_m/2$ is approximately equal to $\phi_m/2$, it follows that the average cutting arc length is

$$(R\phi_m)_{\text{avg}} = 4r/\pi \ldots\ldots\ldots\ldots\ldots [20]$$

Now that we have in Equation [20] the average arc length we can use it in Equation [18] to obtain the average drag distance when cutting off fixed round stock

$$\underline{x_{d0\,\text{avg}} = 2r/\pi} \ldots\ldots\ldots\ldots\ldots\ldots [21]$$

Rotated Work (Forward Rotation). For round stock rotated in the forward direction, we can go through the same steps as those used for fixed work by commencing with Equation [6], dividing by $R\phi$ to obtain the work cut per wheel inch per pass

$$\frac{\Delta A_f}{R\phi} = \frac{C\omega}{\Omega}(1 - \cos\phi)$$

The instantaneous cut depth is the derivative

$$\frac{d}{d(R\phi)}\left(\frac{\Delta A_f}{R\phi}\right) = \frac{C\omega}{R\Omega}\sin\phi = \frac{N}{R}\sin\phi$$

From Fig. 7, the drag distance is

$$x_{df} = \frac{1}{\int_0^{\phi m} N\sin\phi d\phi}\int_0^{\phi m} R(\phi_m - \phi)N\sin\phi\,d\phi$$

$$x_{df} = \frac{R\,\phi_m{}^3}{6\,(1 - \cos\phi_m)} \ldots\ldots\ldots\ldots\ldots [22]$$

By substituting $\cos \phi_m = 1 - \phi_m^2/2$, we obtain as a close enough approximation

$$x_{df} = R\phi_m/3 \qquad [23]$$

(Compare with Equation [18]).

The second step for forward rotation is to find the average cutting arc $\phi_{m\ \mathrm{avg}}$ when cutting off round stock. From Equation [4a] we can write

$$\cos \phi_m = 1 - \frac{\dot{A}}{R\omega(R + r_2)} \qquad [24]$$

Since $\cos \phi_m = 1 - \phi_m^2/2$ (approx) Equation [24] may be converted to the form

$$\phi_m = \sqrt{\frac{2\dot{A}}{R\omega}}\sqrt{\frac{1}{R + r_2}} \qquad [25]$$

The average value of the arc ϕ_m is then

$$\phi_{m\ \mathrm{avg}} = \frac{1}{r_1}\sqrt{\frac{2\dot{A}}{R\omega}}\int_0^{r_1}(R + r_2)^{-1/2}\,dr_2$$

$$\phi_{m\ \mathrm{avg}} = \frac{2}{r_1}\sqrt{\frac{2\dot{A}}{R\omega}}\left(\sqrt{R + r_1} - \sqrt{R}\right) \qquad [26]$$

On substituting the average arc from Equation [26] in Equation [23], we obtain the average drag distance when cutting off round stock of radius r_1 with forward rotation

$$x_{df\ \mathrm{avg}} = \frac{2}{3r_1}\sqrt{\frac{2R\dot{A}}{\omega}}\left(\sqrt{R + r_1} - \sqrt{R}\right) \qquad [27]$$

Rotated Work (*Backward Rotation*). The average drag distance when cutting off round stock with backward rotation can be found by steps similar to those which have just been completed for zero and forward rotation of the work. We commence with Equation [9] and divide by $R\phi$ to obtain the total cut per wheel inch per pass

$$\frac{\Delta A_b}{R\phi} = \frac{C\omega}{2\Omega}\left(\phi^2 - 3\phi_m\phi + 3\,\phi_m^2\right) \qquad [28]$$

The instantaneous cut depth is

$$\frac{1}{R}\cdot\frac{d}{d\phi}\left(\frac{\Delta A_b}{R\phi}\right) = \frac{C\omega}{2\Omega R}\left(2\,\phi - 3\,\phi_m\right) \qquad [29]$$

The average distance the metal is carried is

$$x_{db} = \frac{1}{\dfrac{C\omega}{2\Omega}\displaystyle\int_0^{\phi_m}(2\,\phi - 3\phi_m)d\phi} \times \int_0^{\phi_m}\frac{C\omega R}{2\Omega}(\phi_m - \phi)\,(2\phi - 3\phi_m)d\phi$$

$$x_{db} = (7/12)\,R\phi_m \qquad [30]$$

Equation [30] for backward rotation should be compared with [18] and [23], for zero and forward rotations of work, respectively.

For backward rotation, the average cutting arc $\phi_{m\ \mathrm{avg}}$ is the same as for forward rotation, Equation [26]. On substituting in Equation [30], we find that the average drag distance while cutting off round stock with backward rotation is the following

$$x_{db\ \mathrm{avg}} = \frac{7}{6r_1}\sqrt{\frac{2R\dot{A}}{\omega}}\left(\sqrt{R + r_1} - \sqrt{R}\right) \qquad [31]$$

To illustrate the numerical behavior of Equations [21], [27], and [31], the same conditions have been used as were used before, namely, a 24-in. wheel at 2100 rpm cutting off 6-in. round stock. Fig. 8 illustrates the influence of work rotation and cutting rate on the average drag distance.

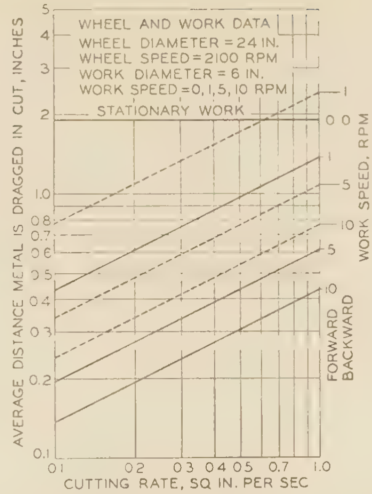

FIG. 8 COMPUTED CURVES SHOWING AVERAGE DISTANCE METAL IS DRAGGED IN CUTS THROUGH CYLINDRICAL WORK AS FUNCTIONS OF CUTTING RATE

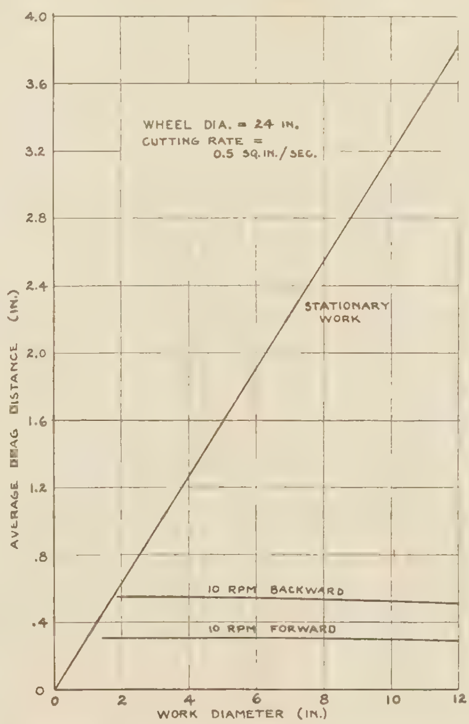

FIG. 9 COMPUTED CURVES SHOWING INFLUENCE OF WORK DIAMETER ON AVERAGE DRAG DISTANCE

Evidently even backward rotation at moderate speeds is an improvement over fixed work, while the advantage gained by forward rotation is still greater. For example, with a cutting rate of 0.3 sq in. per sec, the average drag distances are as follows:

	Inches	Relative
Work fixed	1.90	7.9
10 Rpm backward	0.42	1.75
10 Rpm forward	0.24	1.0

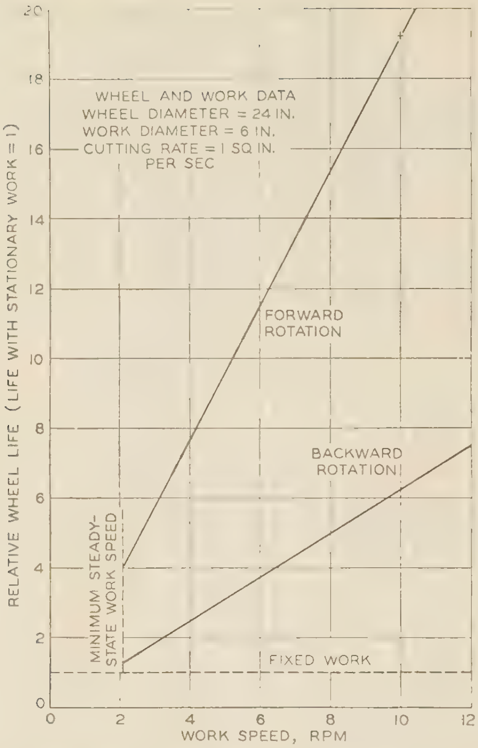

Fig. 10 Computed Curves Showing Estimates of Relative Wheel Life as Influenced Only by Dragging of Metal Through Cut as Functions of Work Speed

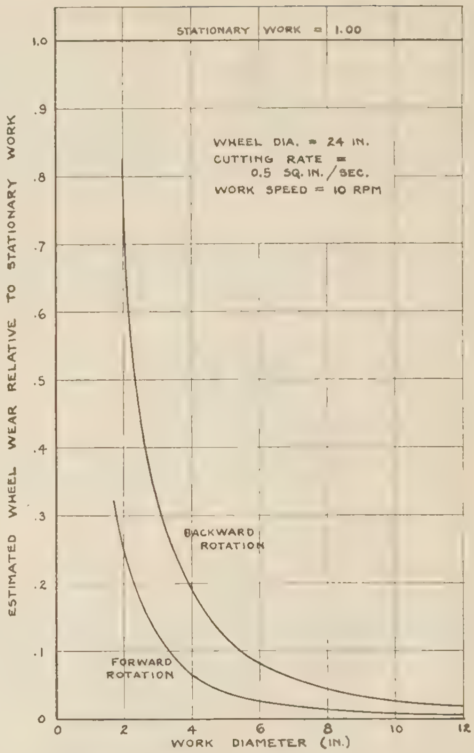

Fig. 11 Computed Curves Showing Estimates of Relative Wheel Wear as Influenced Only by Dragging of Metal Through Cut as Functions of Work Diameter

The influence of work diameter on the average drag distance is illustrated in Fig. 9. The drag distance is almost the same for all work diameters, when the work is rotating.

By returning to the earlier discussion of wheel life and Equation [15], we can make some estimates of how rotating the work might influence the relative number of cuts which could be made in the life of a wheel.

If the ratio of cutting rate to wearing rate were proportional to $(\sigma/x_d)^2$ the relative wheel life would be illustrated by Fig. 10.[4] Actually, there is serious wheel wear even when the drag distance approaches zero, as in light finish-grinding of rapidly rotated work in a lathe. This represents an additional element of wear, which is added to the foregoing.

The influence of work diameter upon the wheel wear, resulting from drag, would then be as shown in Fig. 11, which gives relative wheel wear.

Interpretation

All of the equations that are given, except Equation [15], are believed to be rigorously correct. They are based only on the pure geometry of cutting-wheel action. No physical theories are involved except the concept of drag distance, and, consequently, it should be safe to use them with confidence in planning and in carrying out such experimental work as is necessary to provide detailed information about the action of an abrasive cutoff wheel.

The prospective value of this analytical investigation lies in its prediction that a large and valuable increase in wheel life is attainable by operating it to best advantage. Final determination of the best conditions can only be made by means of a series of tests, planned and interpreted in the light of the theoretical knowledge which we now possess.

Most important would be an experimental determination of the ratio of cutting rate to wheel wearing rate in terms of the wheel porosity and the drag distance to establish the function represented by Equation [15]. Curves which could then be drawn to correspond with Figs. 10 and 11 would show the true importance of drag distance upon wheel life.

Acknowledgment

The author acknowledges the co-operation of the Walker-Turner Co., Inc., Plainfield, New Jersey. Through their Mr. G. L. Dannehower, they provided facilities and assistance with experimental work.

Discussion

P. L. Alger.[5] Mr. Heinz' paper is particularly interesting to the writer, because of the difficulties we have experienced in securing uniform surface conditions without burring in the grinding of laminated air-gap surfaces of electric motors. It is recognized that the author's theory applies primarily to grinding operations, where there is a large arc of contact, but the writer would like to inquire whether it may have any bearing also on the problem mentioned.

In any case, it should be interesting to correlate the smoothness or other qualities of the ground surface with the factors of speed, direction of rotation, and drag distance. If the drag distance is

[4] The "minimum steady-state work speed" indicated is that work speed at which the cut depth would extend entirely to the center of the work.

[5] Apparatus Design Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

of the importance indicated, in determining the useful life of the cutting wheel, it should also be very important in determining the scoring or smoothness of the ground surface. In the latter case, there may be important differences in the surface obtained, even when the arc of contact is very small, depending upon the factors which the author has analyzed.

A. H. Dall.[6] Mr. Heinz has presented a novel approach to the geometry of the grinding process. This new approach attaches great significance to the effect on the wheel due to interference with the action of the grain by the presence of the chip *after* its removal from the work but before its release from the wheel. Previous theories emphasized the effect on the wheel due to the maximum force on the grain *while* the chip was being formed. Thus, in the Alden theory,[7] an expression for the maximum chip thickness was derived. Guest, in his theory,[8] also derives an expression for the maximum chip thickness. Both of these theories were based on the hypothesis that the grains break away from the wheel under the influence of the maximum cutting force, which in turn is a function of the maximum chip thickness. Alden proposed that this cutting force was proportional to the maximum chip thickness, while Guest proposed that the maximum cutting force was proportional to the square of the chip thickness. Their equations, using the author's nomenclature, are as follows

$$F_{\max} \propto \frac{v^2}{V^2N^2} \times \frac{r + R}{Rr} t \quad \text{(Guest)}$$

$$F_{\max} \propto \frac{v}{VN} (\sin \phi + \theta) \quad \text{(Alden)}$$

where

v = peripheral speed of work
V = peripheral speed of wheel
N = number of grains per inch of wheel periphery
r = radius of work (average)
R = radius of wheel
t = one-half cutting depth $\frac{r_1 - r_2}{2}$

ϕ and θ are angles as shown in Fig. 1 of the paper.

Alden's equation can be reduced to the same terms as Guest's equation by the application of the cosine law and approximate cosine series. Thus

$$F_{\max} \propto \frac{v}{VN} \sqrt{\frac{R + r}{Rr} t}$$

While no experimental quantitative verification of these formulas has ever been presented, it is nevertheless well known that wheels act in a qualitative way according to these equations. Thus, for example, higher work speeds will cause the wheel to appear to be softer, while higher wheel speeds will increase the apparent hardness.

Mr. Hutchinson[9] derives an expression for the average chip thickness. The geometric approach is somewhat similar to that of Fig. 1 in the present paper. Mr. Hutchinson used rolling circles and track circles to indicate the relative motions of wheel and work. The respective curves of cutting paths are shown to be hypotrochoidal and epitrochoidal for the two methods designated "forward" and "backward" cutting, respectively, by Mr. Heinz. Incidentally, Mr. Hutchinson uses the common milling terms for these two methods, i.e., "up-cut" for "forward" and "down-cut" for "backward."

Mr. Hutchinson's approach shows quite clearly the difference between length of chip in backward and forward cutting. Thus it is shown that the chip that is produced by backward cutting gives a slightly shorter chip than that for forward cutting. The volume of the chip for both methods is exactly equal. This is also shown in the author's equations and curves.

The author's interpretation of his Fig. 4 indicates a marked advantage in wheel capacities of the "forward" method over the "backward" method. Since the volume of each chip in both methods of cutting is equal, it is difficult to understand why the crowding of the chip in the pore or void space in front of the grain should cause more interference in the one case than in the other.

Mr. Hutchinson[9] states that heating of the work can be somewhat reduced by changing from forward to backward cutting, thus indicating greater efficiency. Since in backward cutting the chip encounters its greatest interference at the beginning of the cut, superficial dullness of the grain would have less effect than in forward cutting, where the dullness might cause sliding and consequent generation of heat. The backward method of cutting is exemplified by the centerless grinder while all center-type grinders use the forward method. Observation of hundreds of practical grinding operations by both methods, however, indicates no apparent difference in capacity between the two methods.

Many tests have been made by the writer's company in which accurate methods were used to measure grinding-wheel wear. It was found that, in general, the ratio of metal-cutting rate to grinding-wheel wearing rate was high for hard wheels (small pores) and low for soft wheels (large pores). However, the smaller wear (breaking away of grains) on hard wheels caused dullness of the grains and consequent rise in power, until cutting became impractical on the particular steel being cut. On the other hand, the very soft wheels wore excessively. Unfortunately, the increase in wheel porosity is usually accompanied by a decrease in bond strength.

From the results of these tests, a criterion of wheel efficiency was set up, based on the economics of grinding. Thus the cost of power plus the cost of wheel (both loss by wear and loss by truing) per hour showed that the most economical wheel in any given bond series was neither the softest nor the hardest wheels but some wheel between these limits.

The conclusion was reached that the optimum wheel was one in which the grains broke away as they became dulled to a point where their usefulness as cutting tools ended, thus permitting new grains to take up the cutting burden. A proof that this action is probably correct was shown by the power versus time curves. In the optimum wheels, these curves leveled off, giving rise to the conclusion that the average number of dull and sharp grains remained substantially constant during sustained operation.

The author is to be commended for his excellent mathematical treatment of his physical concept of the grinding operation. It is possible that a combination of the force theories and the interference theory may bring the physical constants in correct perspective with respect to the wheel action. However, in the light of the results of extensive investigation, it does not appear that the interference theory in itself is sufficient to establish a criterion for grinding-wheel performance.

K. F. Whitcomb.[10] From a geometrical standpoint, this paper is excellent.

[6] Engineer, Research Department, The Cincinnati Milling Machine Company, Cincinnati, Ohio.

[7] "Operation of Grinding Wheels in Machine Grinding," by G. I. Alden, Trans. A.S.M.E., vol. 36, 1914, pp. 451–460.

[8] "A Theory of Work Speeds in Grinding," by J. J. Guest, presented before the British Association for the Advancement of Science, 1914.

[9] "Do We Understand the Grinding Process?" by R. V. Hutchinson, *S.A.E. Journal*, vol. 42, Transactions Section, 1938, pp. 89–100.

[10] Research Laboratories, Norton Company, Worcester, Mass.

The author has presented three factors which, he argues, influence wheel wear and material removed when cutting off stock with an abrasive wheel. They are:

1 Unit contact force.
2 Average carry of the metal or drag distance.
3 Porosity of wheel.

We agree with the author that the unit contact force will influence both wheel wear and material removed.

The author's second factor, that of average drag distance of metal chips in the cut and its effect on wheel wear, is a very interesting speculation. Is the greater wear, as the drag distance increases, due to greater heating of the bond at the point of contact which in turn causes more wheel wear, or is it due to bending and impact against loose chips? Certainly the metal chips with larger-diameter stock are dragged a greater distance and each point on the wheel face will be in contact with the hot metal bar being cut for a longer period of time before it clears the work. It seems as if both factors may be important.

Recent cutoff wheel testing by the writer's company has given the following results: Keeping type of wheel, thickness of wheel, speed of wheel, and time per cut constant, the cut was made dry in one case while, the cutting was done wet in the second case; the wheel wear when cutting wet was half what it was when cutting dry. Probably the water tended to keep the bond cooler and thus stronger and finally bolstered it up to withstand the impact and bending forces mentioned by the author. This indicates that heating of the wheel face while in contact with the work is also a very important factor influencing wheel wear.

Relative to the author's third point pertaining to porosity of the wheel as a whole and its effect upon wheel wear and material removed, we submit the following:

It is our opinion that the porosity of the wheel face is very important and is related to rate of cut. However, with organic bonded wheels, of which cutoff wheels are an example, we have found that two wheels made with the same volume of bond and abrasive, but different types of bonds, may have entirely different porosities on the wheel faces once grinding has been done. We have proved quite conclusively for certain types of grinding that, with lower areas of contact between wheel and work, we obtain higher rates of cut. Open wheel faces or faces with high porosity can be easily obtained by using less bond in the wheel, that is, making a softer wheel. The problem comes in obtaining bonds with the correct heat resistance and proper physical properties at elevated temperatures so that wheel wear may be kept to a minimum while retaining the porosity on the wheel face.

The author seems to have assumed that as soon as the wheel face clears the work the chips held in the pores are released and the open pores are again ready to take in new chips formed. Our experience indicates that this is not true but that some pores relieve the chips formed, while in other cases much of the pore space remains filled and metal loading results.

We have found this metal loading to be a very important factor on influencing wheel wear and material removed. When certain fillers are introduced into the bonds or pores, loading has been reduced and with it heat has been removed from the wheel face. This has resulted in lower wheel-wear values and the same rates of cut. Finally the ratio of rate of cut to wheel wear has been increased.

G. L. Dannehower.[11] Rapid advances have been made during the last decade both in the manufacture of abrasive cutoff wheels and in their application to cutting problems. In the very near future we shall see great improvements in the cutting of alloy steels and nonferrous metals and plastics with vastly greater economy, less effort, and a higher degree of accuracy. The author has provided information that is highly important in our search for progress in this particular work. Certain things that we were imagining as going on at the arc of contact are now more accurately known.

[11] Walker-Turner Company, Inc., Plainfield, N. J.

Experimental work conducted by the writer has confirmed some of the basic equations in the paper. Furthermore some results of entirely new significance have been obtained, thanks to the methods for analyzing the experimental data which follow from this paper. The author's fundamental analysis has already proved its practical value in guiding us in studying abrasive-wheel operation.

Accurate knowledge of performance will not only guide the wheel manufacturer in improving the very structure and characteristics of thin cutoff wheels, it will also assist the designer in improving his machines. It will act as a guide to the machine operator in obtaining production records which now seem beyond attainment. Rapidly moving abrasive particles properly directed and controlled under correct mathematical conditions will achieve remarkable results in metal-cutting service.

Close attention and thorough study of such analytical investigations as that reported by the author will assist us in accomplishing results that in the art of metal cutting far surpass any hitherto contemplated. To this end we too can use mathematics properly, as the very important tool that it is. With its guidance we can accomplish more in less time than can be realized by trial-and-error methods. We are not going to be content in our accomplishments with thin abrasive cutoff wheels until we do more than just "scratch the surface."

Author's Closure

The paper under discussion is directed primarily toward the influence of various working conditions upon a cutoff wheel and practically not at all toward their effects upon the resulting work-surface finish. In the realm of cutoff wheel service, the arcs of contact between wheel and work are relatively large, and drag distances are great. In surface grinding, on the other hand, feed rates are so small that arcs of contact and drag distances approach the infinitesimal. Also, the cutting rates are so low that wheel pores are probably far from being packed after a single pass through the work. Consequently, it seems doubtful that reliable predictions regarding wheel operation for best surface finish could be drawn from this particular analysis. It seems that more light might be thrown upon Mr. Alger's problem by analyses of individual grain action, such as those referred to by Mr. Dall.

In his discussion Mr. Dall mentions hundreds of practical grinding tests with both forward and backward rotation which indicated no apparent difference in capacity between the two methods. This was because the tests he mentioned were all with cutting rates which were low as compared with those which are common in cutting-off work. Consequently, arcs of contact, drag distances, and packing of the wheels were probably all so small that differences among them of several hundred per cent would have had negligible effects. In other words, surface-grinding work is conducted in a realm of operation completely outside the realm with which the analysis in the present paper deals.

The author, like Mr. Dall, once found difficulty in arriving at a satisfactory physical explanation for the inescapable significance of Fig. 4 of the paper. However, the situation no longer seems puzzling. Fig. 4 shows that a wheel of 0.0002-in. porosity, for example, would cut smoothly into the work used for computation at a rate of 0.4 sq in. per sec when the work rotates forwardly. It says, however, that if that same wheel be fed at the same rate into the same work rotating backwardly, the pores of the wheel would be packed instantaneously on first contacting the work.

The reason is (see Fig. 1 of the paper) that each incremental arc of wheel rim enters the wedge-shaped slice of metal from the thick end. No grains ahead of our incremental arc have pared away any of that thick portion of the wedge. Consequently, a great depth of metal must be crowded into the pores of the wheel. If this depth be greater than the pores can hold, the wheel will momentarily stop cutting until the loaded part has passed out of the arc of contact, at which time an empty section of the wheel can take another bite. Thus, the wheel will alternately cut and refuse to cut, with an alternating frequency measurable in cycles per second. This action results in the rough noisy performance which is commonly termed "pounding." It is well known that such action quickly knocks a wheel to pieces.

With forward rotation, each increment of arc on the wheel rim enters the work from the thin end of the wedge, preceded by other similar increments ahead of it which have pared out small portions from the arc of contact. Thus, by the time any particular incremental arc gets to the thick end of the wedge, its predecessors have loaded themselves with metal and there is only a relatively small amount remaining for the final increment to take away. Under these conditions, the wheel can cut smoothly and continuously without packing its pores. This also is confirmed by experimental evidence.

Mr. Dall is undoubtedly correct in his statement that the interference theory in itself does not appear "sufficient to establish a criterion for grinding-wheel performance." Interference is only one of many influential factors, but it is one which is both important and subject to analysis. Fig. 12 of this closure shows that the wear factor[12] is less at a certain drag distance than it is at any other. At high drag distances the interference probably rules. At low drag distances interference is small, while other considerations become increasingly serious. The grain-loading theories, reviewed by Mr. Dall, probably are important in this region, where metal is being removed at the same rate, but by shorter peripheral arcs of wheel.

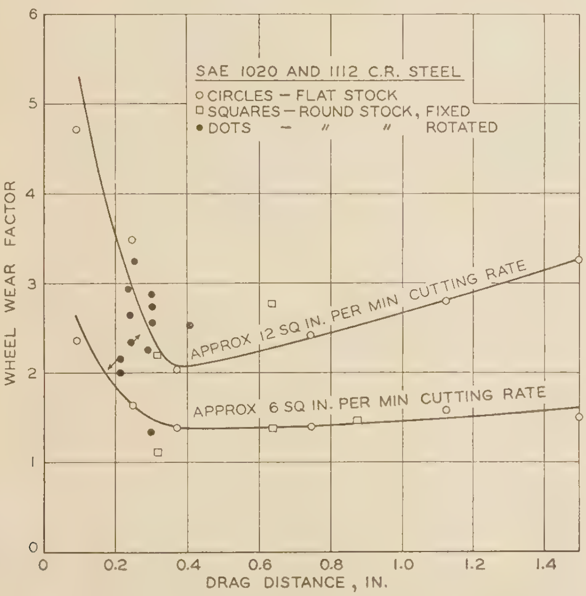

Fig. 12 Experimental Correlation Between Wheel-Wear Factor and Average Drag Distance Under Various Cutting Conditions

The curves of Fig. 12 were plotted from a large number of experimental cuts through round stock, both fixed and rotating at various speeds, and through flat stock. In spite of known experimental deviations, these curves show a definite correlation between the wear factor and the drag distance under numerous different kinds of cutting conditions.

As the theory predicts, chip interference causes the wheel wear to increase more rapidly at a cutting rate of 12 sq in. per min than at 6 sq in. per min. The minimum wear factor seems to occur at the same drag distance regardless of cutting rate.

At the left the curves rise steeply as the drag distance becomes small and individual grain loading increases. There is no evidence as to where they would go if carried still further toward the left, into the region in which surface-grinding performance lies. The experience referred to by Mr. Dall has dealt with operation in this region. His tests may have been confined to lower cutting rates than these and to drag distances in the order of 0.01 in. It should be noted that Mr. Dall has published a detailed analysis of the several grinding theories.[13]

Mr. Whitcomb discusses additional variables which affect cutting action: namely, heating of the wheel, heating of the work, metal loading of the wheel pores, and comparisons of wet versus dry grinding. The author regards the first three as departures from constancy in the "constants" of his analysis. Heating influences the bond and thus the effective grade of the wheel. Heating of the work affects its cutting characteristics. Metal-loading changes the wheel porosity. All of these factors have been dealt with as though they are constant, even though they are known not to be so. The author can imagine no manner in which variations in them can be introduced mathematically. But it is not necessary to do so in order to profit from rigorous analysis of the ruling phenomena.

Wet grinding involves yet other effects which seem unapproachable mathematically, i.e., the hydraulic forces which act upon the chips, lubrication of surfaces on the work, on the chips, and in the wheel pores, etc. These problems, like the foregoing, are best worked out experimentally, upon the basis of a mathematically guided plan.

[12] "Wear factor" is the ratio of square inches cross section of metal cut to square inches of wheel worn away. It is the inverse of the service factor.

[13] "A Review of the Grinding Theories," by Albert H. Dall, *Modern Machine Shop*, vol. 12, no. 6, November, 1939, pp. 92–112.

Pulverized Coal for Forge Furnaces

By R. B. ENGDAHL[1] and F. E. GRAVES,[2] COLUMBUS, OHIO

In the course of a survey to determine possibilities for the use of coal in the metallurgical industries, conducted for Bituminous Coal Research, Inc., at Battelle Memorial Institute, studies were made of the application of pulverized coal to forge furnaces. The results obtained and data compiled from the investigation of the action of pulverized coal in a small forge furnace in the laboratory are given in this paper. The conclusion is reached that pulverized coal is entirely suitable for use on many types and sizes of forging furnaces; that it has the same advantage of fluidity as gas and oil; and that it gives a flame of high emissivity which provides maximum heat transfer by radiation. Of particular importance is the fact that for most sections of the country it is the most economical fuel to use for this purpose.

The shaping of metals when heated to a temperature at which they are plastic is an important technique in modern industry, notwithstanding its antiquity. From the hand methods of the prehistoric blacksmith the method has been developed and improved until today forgings go into the construction of practically every modern machine, be it automobile or tank, turbine or plane. Between 5 and 7 per cent (1)[3] of the weight of the modern automobile is composed of forgings; most of these are at vital load-carrying points. The total weight of forgings produced annually in this country averaged 503,000 tons from 1924 to 1939 (2). In 1941 this amount increased to approximately 1,530,000 tons. Fuel consumed in the period 1924-1939 accounted for 3 to 4 per cent of the cost of the finished product, roughly 3 to 4 million dollars annually.

In connection with a survey of the metallurgical industries to determine the possibility for the application of coal, which is being made by Battelle Memorial Institute for Bituminous Coal Research, Inc., the forging industry has been reviewed. This paper presents the results of this survey and data obtained in an investigation of the performance of pulverized coal in a small forge furnace.

Design of Forge Furnace

The ideal toward which all furnace design should be directed is to provide clean stock uniformly heated to a predetermined temperature at just the rate demanded by the forging equipment. Furnaces now in use satisfy these requirements only partially because of limitations imposed by the size of forgings, burners, fuels, refractories, and production demands. Faulty design which is almost universal has also been pointed out as being responsible for many of the shortcomings of forgings (3), but that problem requires treatment beyond the scope of this paper. Before considering the effect of the other variables, typical forge urnaces will be described.

Furnaces for handling forgings whose weight ranges from less than 1 lb to 40 tons (4) naturally vary greatly. Large forgings, of which only a few pieces at a time may be fabricated, are heated in batch furnaces. Such forgings are usually of heavy cross section and require very slow heating if the center is to be brought to forging temperature without overheating the surface.

Smaller forgings, and 70 per cent of all forgings weigh 15 lb or less (2), are heated either in batch or continuous furnaces. These are called drop forgings because of the drop type of hammer often used in their fabrication. They are usually made in large quantities; duplicate sizes and shapes being obtained by means of shaped dies into which the plastic metal is forced to flow under repeated blows. Continuous furnaces for this class of work are either the pusher or rotary-hearth types, but a large part of small-production forgings are heated in a simple batch furnaces. Either the pusher or rotary types employ mechanical means for moving the work through various heating zones at a controlled rate to supply properly heated stock at just the rate demanded by the hammer. Much more common is the box type in which the work is manually charged through a door opening or through a narrow, uncovered slot; hence the name "slot furnace." Hearth dimensions for this common type of furnace range from 1.5 × 3.5 to 3 × 5 ft.

In an effort to have maximum contact of the gases with the work, no special provision is made in these small furnaces for removal of the products of combustion; instead, they flow out of the slot which is partly filled with the projecting ends of stock. This high-velocity stream of hot gases is thus directed at the operator who is partly protected by a series of air or steam jets so aimed as to deflect the flames upward in a short stack formed by a radiation shield supported some distance in front of the furnace wall. These measures are only partially effective, and redesign for increased operator comfort is much needed.

Frequently there is no automatic control of temperature. The operator removes units as they reach forging temperature which he judges from appearance of the piece. A number of units are in the furnaces at one time and new ones are added as hot ones are removed. Thus, this might be considered a manually controlled and operated continuous type of furnace. However, the inevitable defects of manual control and operation are fuel and material waste owing to poor control and nonuniformity of product. It is to be hoped that in the interest of fuel economy, operator comfort, and more satisfactory and faster production, the continuous thermostatically controlled furnace will find much greater use in the near future.

Requirements of Forge Furnaces

Uniformity of Heating. Uniform heating of the steel throughout the section, in order to avoid unequal flow between the metal near the surface and that at the core, is well recognized as being desirable; usually this condition is fairly easy of attainment. Complete uniformity during heating is possible only with electric-induction or resistance heating. In furnaces fired by oil, gas, or coal, reasonably uniform temperature in the stock is attained by allowing it to remain in the furnace for a sufficient length of time. This time requirement must be kept as low as possible in the interest of fast production. If the size of stock is increased, the time for heating must be increased to allow the temperature to equalize within the bar. The forging trade has developed empirical rules for this. For example: "Steel bars for drop-forge work can be heated at the rate of 5 to 15 min per in. of diam." (5).

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[1] Research Engineer, Battelle Memorial Institute. Mem. A.S.M.E.

[2] Research Engineer, Battelle Memorial Institute.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Fuels Division and presented at the Semi-Annual Meeting, Cleveland, Ohio, June 8–10, 1942, of The American Society of Mechanical Engineers.

Although this and similar rules indicate the time required to achieve uniformity of temperature in work heated in the usual forge furnace, they are often interpreted to specify in addition the maximum safe heating rates. It is implied that at higher rates the steel will be injured by the unequal expansion which accompanies unequal temperature. However, Murphy and Jominy (6) have shown that much higher rates can be safely employed. In a small carbon-resistor furnace initially at 2870 to 3000 F, 1.5-in. round bars were heated from 70 F to 2200 F at the surface at the rate of 3.3 min per in. of thickness; when the surface reached 2200 F the core was only 24 F cooler. No damage was suffered by the sample. During the time that the sample was heating, the furnace temperature was allowed to drop to 2400 F in order to avoid overheating the sample. This last condition serves to emphasize that the real danger in rapid heating is that of overheating or "burning" the steel, if such heating is attained by means of high furnace temperatures rather than by increasing the heat-transfer coefficients.

In fuel-fired furnaces, the work receives energy by radiation and convection from its surroundings which are at higher temperature. In attempting to attain uniform temperature rapidly, the usual method is to increase the heat transfer by raising the furnace temperature. This may be carried to the point where very rapid oxidation of the steel can occur; hence if the work is not removed instantly when it has reached forging temperature, this oxidation, known as "burning," may, and often does, ruin the steel.

An alternative method for securing uniform and rapid heating is to increase the heat-transfer coefficients. At forging temperatures, the rate of heat transfer by convection is small relative to that by radiation. Measurements made by Schack (7) in a rolling-mill furnace showed that convection accounted for only 12 per cent of the total heat transfer. Calculations (8, 9) usually based on the limited work by Jurges show that the convection coefficient accounts for only 5 or 6 per cent of the total in particular cases. Hence with radiation playing the predominant role, any important increase in the total heat transfer must come from an increase in the radiation coefficient. The luminous flame is used to achieve this end, although Schack and Sherman (10) have shown that what increase in heat transfer is obtained in this way is limited because of the increased absorption of wall radiation by the flame as the luminosity of the flame is increased.

The pulverized-coal flame has been shown to emit at a rate even greater than that of the luminous natural-gas flame (10). This condition gives rise to the common remark that the "heat from a pulverized-coal flame is softer and more penetrating." What is meant is that instead of securing maximum heat transfer by lashing the stock with visibly high-temperature flame, heating is effected by radiation from the ash, coal, and coke particles without such visual evidence of the active heat transfer.

High heat-transfer rates have already been stated to permit uniformity of heat in a minimum of time, thus aiding rapid production. Reduction of the time required for heating is important also in that it reduces scaling.

Effect of Furnace Atmosphere. The formation of oxide or scale not only causes the loss of a certain amount of steel but also requires extra labor to remove it. If not removed, it pits the forging and wears the dies. Furthermore, scale forms slag with the refractories which erodes the furnace walls and hearth. Hence its elimination or reduction is desirable.

The variables affecting scaling are time, temperature, and atmosphere. The first two have been shown to have definite effects; this indicates that the time for heating should be kept to a minimum, and the temperature of heating should be kept as low as is consistent with proper flow of the metal during forging. The effect of furnace atmosphere is not so definite.

When steel is heated to forging temperatures in the atmosphere of a direct-fired furnace, the scale is formed on the steel by oxidation by the free oxygen, water vapor, carbon dioxide, and sulphur dioxide which result from combustion of the fuel. The scaling effect of each of these flue-gas constituents has been studied by several investigators. Fig. 1 from Jominy and Murphy (6) shows that water vapor has just as strong a scaling effect as free oxygen at temperatures below 2400 F, and that carbon dioxide will cause more scaling than will air. Sulphur dioxide was also found to have a very strong scaling effect when present to the extent of 0.1 to 0.2 per cent.

However, no information is available on the relative oxidizing effect of these flue-gas constituents when they are combined in the proportions common to the atmosphere of a direct-fired forge furnace. It would be wrong to try to predict exactly the amount of scaling that would be caused by a combination of gases whose individual scaling actions are known; as Jominy and Murphy state, the possibility of variation in the physical and chemical properties of the scale produced makes such a prediction unwise. Nevertheless, a knowledge of the scaling tendencies of free oxygen, water vapor, carbon dioxide, and sulphur dioxide does make it possible to predict the general scaling effect of a furnace atmosphere containing these gases in various proportions.

An atmosphere that is considered reducing, from a standpoint of combustion, will slow down scaling; but the water vapor and carbon dioxide in the products of even very incomplete combustion make flue gases oxidizing to hot steel. The formation of scale will be impeded by anything that reduces the amount of oxygen present as free oxygen, water vapor, carbon dioxide, or sulphur dioxide. Bullens (11) states that 75 per cent of the scale can be eliminated in an atmosphere containing 3 per cent carbon monoxide and no free oxygen, but that it requires 15 per cent carbon monoxide in the flue gas completely to eliminate scaling. However, such a reducing atmosphere would cause excessive decarburization because both carbon dioxide and water vapor must be absent if decarburization is to be prevented.

When steel is heated to forging temperatures in a direct-fired furnace, where it is impossible to eliminate both decarburization and scaling, the effort to avoid scaling will usually result in the greater disadvantage of decarburization. In heating steel for forging, it is usually best to aim at producing a light scale that is not too objectionable for hot-working but which is not accompanied by excessive decarburization.

Table 1 shows the composition of the flue gases from the combustion of natural gas, oil, and coal, with varying amounts of excess air. To achieve the same combustion efficiency in a small forge furnace, it is usually necessary to burn pulverized coal with a slightly larger amount of excess air than is required with oil and a much larger amount of excess air than is required with gas. A comparison of the composition of the flue gas resulting from burning coal, oil, or gas with no excess air will show that, while there is considerably less water vapor in the products of combustion of coal than with oil or gas, the atmospheres obtained with all three fuels are definitely oxidizing.

Table 1 and Fig. 1 show that if coal is burned with 30 per cent excess air, oil with 20 per cent excess air, and gas with no excess air, the scaling effect of the higher proportions of free oxygen and carbon dioxide in the flue gas from coal is probably no greater than the scaling effect of the higher proportion of water vapor in the flue gases from oil and gas, and that each of the atmospheres produced has approximately the same over-all scaling effect. Considering, however, the firing of any one of these fuels, the amount of scaling may be somewhat reduced by reducing the proportion of free oxygen in the flue gas to such an extent that combustion is not complete. Reducing the proportion of free oxygen in the flue gas will increase the proportion of both carbon

TABLE 1 THEORETICAL FLUE-GAS COMPOSITION FROM COMPLETE COMBUSTION OF COAL, NATURAL GAS, AND OIL

Fuel	Excess air, per cent	Per cent by volume, dry gas				Per cent by weight, wet gas				
		CO_2	O_2	N_2	SO_2	CO_2	H_2O	O_2	N_2	SO_2
Coal										
Ultimate Analysis:	0	18.5	0.0	81.4	0.1	25.2	4.1	0.0	70.6	0.1
C 81.0 per cent	5	17.6	1.0	81.3	0.1	24.2	3.9	1.0	70.8	0.1
H 5.3 per cent	10	16.8	2.0	81.1	0.1	23.2	3.7	2.0	71.0	0.1
O 8.4 per cent	20	15.4	3.5	81.0	0.1	21.4	3.4	3.6	71.5	0.1
N 1.5 per cent	30	14.1	4.8	81.0	0.1	19.8	3.2	4.9	72.0	0.1
S 0.6 per cent										
Ash 3.2 per cent										
Natural Gas										
Ultimate Analysis:	0	11.6	0.0	88.4	0.0	15.2	11.7	0.0	73.1	0.0
CH_4 87.0 per cent	5	11.0	1.1	87.9	0.0	14.4	11.1	1.0	73.5	0.0
C_2H_6 7.6 per cent	10	10.5	2.1	87.4	0.0	13.9	10.7	2.0	73.4	0.0
C_3H_8 2.7 per cent	20	9.6	3.8	86.6	0.0	12.8	9.8	3.7	73.7	0.0
C_4H_{10} 0.8 per cent	30	8.8	5.2	86.0	0.0	11.9	9.1	5.1	73.9	0.0
N_2 1.9 per cent										
Fuel Oil										
Ultimate Analysis:	0	15.5	0.0	84.5	0.05	20.5	8.2	0.0	71.3	0.05
C 84.0 per cent	5	14.6	1.1	84.3	0.05	19.5	7.8	1.0	71.7	0.04
H 12.7 per cent	10	13.9	2.0	84.1	0.04	18.7	7.5	2.0	71.8	0.04
O 1.2 per cent	20	13.2	3.8	83.0	0.04	17.2	6.9	3.6	72.3	0.04
N 1.7 per cent	30	12.1	5.2	82.7	0.04	15.9	6.4	5.0	72.7	0.04
S 0.4 per cent										

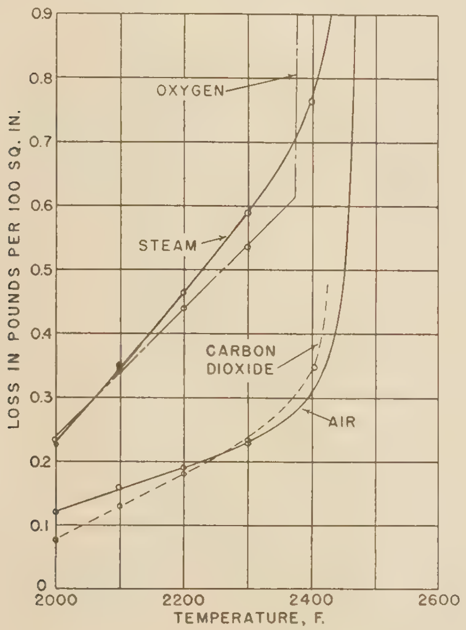


FIG. 1 EFFECT OF VARIATION OF TEMPERATURE ON THE SCALING OF S.A.E. 1015 STEEL
(From Murphy and Jominy.)

dioxide and water vapor, and because the scaling effect of carbon dioxide is not so strong as that of free oxygen, the over-all scaling effect will be lessened.

The problem, therefore, is to reduce the air-fuel ratio only to that point at which the reduction in scaling is not more than offset by the resulting loss in combustion efficiency.

Fuels for Forge Furnaces

In this country the predominant fuels for forge furnaces are now oil and gas. The choice between the two often depends upon availability. This has not always been the case, for coal, the most abundant fuel, has served the needs of the forgers of metal for many years. Fired by hand in lump form, it has been applied to every size and variety of furnace for the heating of forgings. In previous times, the labor costs and lack of control were not important. Development of the principle that coal can be utilized more easily and efficiently in pulverized form led to its application in this state to forges, and again all sizes and types were heated with this fuel. That it was demonstrated to be a practical fuel is testified by the fact that one manufacturer of pulverized-coal equipment listed 360 such furnaces in this country in 1919, and in 1920 there were 690 (12, 13). This is even more striking when it is realized that at that time the application of pulverized coal to boiler furnaces was in its infancy, with many improvements still to be made before the present, smooth-running, efficient plants became possible.

Those were days of scarcity in the fluid-fuel markets, and the economic incentive to reduce costs aided the social incentive for conservation of scarce natural resources for the applications to which they alone are particularly suited.

The discovery of vast new deposits of oil and natural gas in this country led to sharp reductions in the costs of these fuels with consequent enormous increases in their use in all phases of industrial heating including the heating of forgings. The attendant reduction in the use of coal was accompanied by a loss of interest on the part of the equipment manufacturers and the coal industry. As a result, there were no continued efforts toward development and improvement of coal-firing equipment which were essential if coal was to maintain its position in the face of advances in the equipment for using the comparatively low-cost fluid fuels.

That some forge work is sufficiently refined to demand perfectly clean atmospheres is doubtless true, and such cases have been on the increase as new and more sensitive alloys have been developed. Nevertheless, vast quantities of the scarcer fuels have been employed on most of the rougher work without regard to the unnecessary depletion of limited and irreplaceable fuel resources.

Pulverized coal is a suitable fuel for this class of forgings. This is demonstrated by the fact that some installations have been using pulverized coal for steel-heating furnaces for many years, despite the advent of fairly cheap competitive fuels.

Pulverized Coal for Forge Furnaces

Many existing forge furnaces have been converted to pulverized coal without any change in the design of the furnace, while others have been found to demand some changes (14); such changes have usually been to increase length of flame travel or to reduce impingement on refractories. The heat requirements demand combustion rates of 36,000 to 54,000 Btu per cu ft per hr, based on the entire furnace volume. If excessively high localized rates of heat liberation are to be avoided, it is essential that the burner be so designed as to distribute burning through the entire furnace. Theoretical consideration of the time required for combustion or for necessary length of flame travel are of little help in determining burning conditions, as radiation from adjacent refractory, proximity of cold work, intensity of mixing are all influencing to uncertain degrees.

Repeated experience has demonstrated the need for low velocity of admission of the coal to the furnace; that is, with restricted length of flame travel the time for burning is made greatest by reducing the velocity to a minimum. This applies to the entering velocity as well as the velocity through the furnace. However, the burner ports are usually horizontal; and for horizontal entry the velocity must not be so low as to allow the coal to settle out either in the burner or just inside the burner, with

resulting coking and plugging of the port. For the benefit of the furnace designer, the limiting velocity should be stated, but this is difficult since practical values are affected by furnace design and burner arrangement.

For satisfactory transport of pulverized coal, a velocity of 4000 fpm is often specified (15). However, this velocity or even one half this value for a combustion space only 3 to 4 ft long is too high. Flame impingement on the refractories and discharge of the particles from the furnace before burning is completed are inevitable. Furthermore, it is found that satisfactory operation demands inlet velocities of 1500 fpm or less. As settling or segregation occurs at these low velocities, it is necessary so to locate the inlet pipe with respect to the surrounding refractories that the particles will be burned or will have moved past near-by horizontal surfaces before they have time to fall and adhere, thus forming and building up coke. Uniform fineness of coal can be of great help here, since the finer the particle, the greater the tendency to float and resist segregating and settling forces.

After the coal and air are in the chamber, the available space for burning must be well utilized in order to achieve the necessarily high combustion rates. Uniform fineness is of aid as it is in transporting the coal, for fine particles ignite more readily (16, 17) and present more surface for reaction with oxygen. Still the most important factor here is proper distribution of the coal and air in the chamber. This may be attained by means of elaborate mixing burners as are used in boiler furnaces. However, where space is limited, the velocities necessary to produce thorough mixing are apt to cause impingement of the coal and ash on the refractories. Many attempts in this direction in the past (18) have resulted in reversion to a simple straight pipe delivering the coal at as low a velocity as is consistent with satisfactory transport. The point is indicated by the remark by one installer: "We just breathe the air and coal into the furnace . . . ." (12).

Burner location has generally followed ordinary forge-furnace practice with oil. In the small furnaces, with hearths 3 × 5 ft or less, two burners are often employed at opposite sides with each offset from the center line enough to avoid head-on impingement, but still allowing strong intermixing of the expanding flames as they meet. This also prevents direct impingement of the flames on the refractories. Secondary air brought in by induced or forced draft, usually in an annular space surrounding the coal jet, encloses the expanding flame and provides oxygen for completion of the burning. Furnaces having long and narrow hearths may have a single burner at one end, as in Fig. 2 (19), and

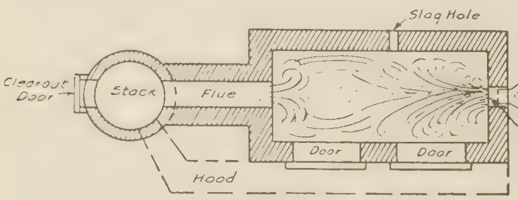


Fig. 2 Small Forge Furnace of the Door Type, Fired With Pulverized Coal
(From Rehfuss, reference 19.)

a flue or stack outlet at the opposite end. The length of hearth is sufficient for satisfactory combustion without any great impingement on refractories. Another variation is to have one burner at each end with a stack outlet in the rear and a slot or door in the front. The working opening is usually a slot or door along a side of the furnace and usually parallel with the flame path.

Although flue gases from standard slot furnaces pollute the atmosphere of the shop, they are seldom removed by hoods or stacks. Natural ventilation is often obtained through large open doors and windows, but forced ventilation would be more satisfactory. When burning coal, the arguments for positive ventilation are bolstered by the necessity for elimination of the fine ash. Many early failures of pulverized coal in forge furnaces were caused either by unwillingness to recognize this fact or by poor and inadequate exhaust systems (20). Furnaces of hearth dimensions 3 × 10 ft and larger can be provided with individual stacks which, since construction details usually result in their being oversize (21), provide cheap, dependable, and controllable means of inducing secondary air. For furnaces smaller than these, space and control considerations make it desirable to install an induced exhaust system with individual hoods connecting to a common breeching. Such an arrangement permits the use of collectors for the fine ash.

The ash which does not pass off with the flue gases either deposits as a thin dust on the surface of the stock, adheres to the walls, or slags off with the scale and refractory gravel commonly used on the hearth. Burner arrangement and furnace design should be such as to deposit this ash with a minimum of impact and so avoid erosion.

Distribution Systems

Large billet, slab, or rail-heating furnaces are often fired directly from individual pulverizer mills with equal success to that attained by direct-fired steam-boiler furnaces. However, to operate these furnaces in conjunction with a power plant or with a number of melting or small forge furnaces, it is desirable to pulverize at a central plant and distribute the coal to the individual furnaces.

Distribution to substation bins from which the coal is fed to the furnaces may be accomplished by means of screw conveyers. For short distances, up to 500 ft, which do not involve obstructions, straight screw conveyers are simple and reliable. However, frequent changes in direction or elevation are awkward to handle with the screw conveyer; hence the systems employing pipes which can be arranged in a manner similar to utility piping, are often more suitable.

Most pulverized materials, when mixed with small amounts of very low-viscosity fluids, form mixtures which demonstrate fluid-like properties. Pulverized coal when mixed with air can be made to flow through standard pipe over distances up to 1500 ft. Three classes of distribution systems utilize this property. They differ chiefly in the amount of air used, air pressure, velocity of mixture, arrangement of piping, and type of prime mover. They are as follows:

1 Compressed-air or blowing system.
2 Pumping system or coal-air-emulsion system.
3 Air-mixture or circulating system.

Fig. 3 shows the mixtures which these systems employ and the relation of bulk density to the air-coal ratio for bituminous coal-air mixtures, at 29.92 in. Hg, 70 F. When compressed to eliminate practically all interparticle spaces, the density of solid coal, air-coal ratio of zero, is approached. At the opposite extreme with the ratio of air to coal approaching infinity the density of clean air is approached. Ideally, an air-coal ratio of about 10-11 is needed for perfect combustion. Actually, in furnaces utilizing pulverized coal, this ratio is about 13-15. Between this condition and that of stored pulverized coal lies the region in which the various transporting systems operate in conjunction with feeders and burners to convert stored coal, with a density of about 35 to 45 lb per cu ft, air-coal ratio of 0.001, to a combustible in scattered furnaces with a density of 0.08 lb per cu ft, air-coal ratio of 13-15.

The inset in Fig. 3, showing a portion of the bulk density-air coal ratio relationship plotted on rectangular co-ordinates, indi-

cates the marked difference in the character of the mixture handled by the various transporting systems. The volume ratio for the pumping and blowing systems is approximately 40 cu ft of air per cu ft of coal, while that for the air-mixture system is 100 times as great. Much information is yet to be gained concerning the properties of fine powders when mixed with low-viscosity fluids. However, indications are that the surface phenomena occurring in these mixtures are such that the velocities necessary to maintain them in the homogeneous state are much less for the low volume ratios than for the high volume ratios. Because the proper velocity is so essential to smooth operation of distribution systems without segregation, plugging, and irregular flow of the coal, further study in this field is greatly needed.

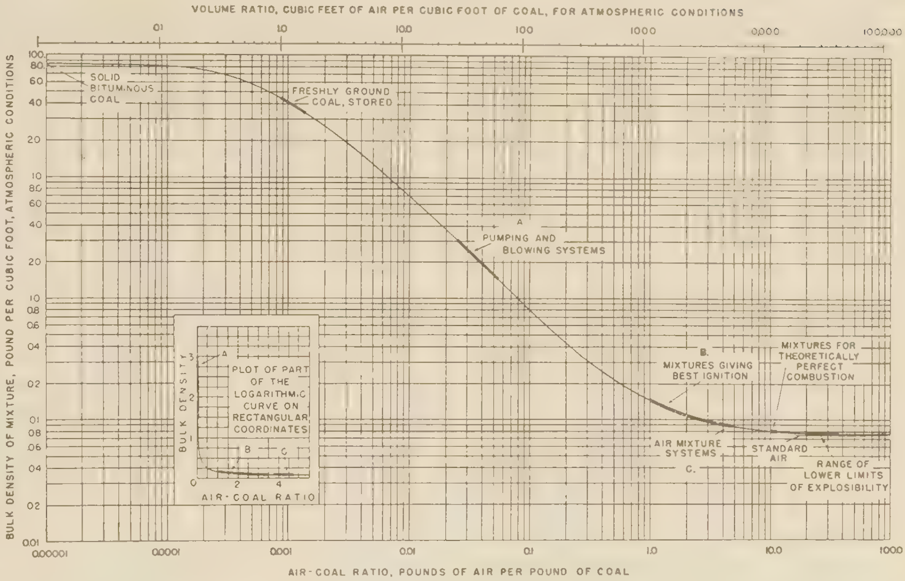


Fig. 3 Bulk Density of Homogeneous Air-Coal Mixtures, 29.92 In. Hg, 70 F

Compressed-Air Transport System. This is often called the blowing system. It employs a tank of 1 to 7 tons capacity, which is intermittently filled by gravity with coal from an overhead bin. Compressed air is admitted to the tank, and the aerated mixture is forced at rates up to 140 tons per hr through a 3 to 5-in. standard pipe to one or more substation bins; separation of air and coal being effected by cyclone separators at the bins. The operator can control the system from the blowing-station control board by means of solenoid-operated valves and bin-level indicators which are connected to the control board from all the substations. Depending on the design of blowing tank, the flow may be in alternate slugs of air and coal or in a uniform mixture.

Pumping System. This is a continuously operating low-velocity system which utilizes either a screw or positive-displacement rotary pump to force the aerated mass of coal through standard pipe. Aeration is attained by means of a compressed-air jet at the pump outlet. At the substation bin, no positive separation of air and coal is necessary, as the rate of flow to the bin is such that the air has time to leave the delivered coal slowly and be vented to the atmosphere without carrying any dust with it. As in the blowing system, the entire system is controlled from the pumping-station switchboard.

In any of the foregoing systems, the mixture is fed at very high rates and does not contain sufficient air for proper ignition. Rapid ignition is to some extent a function of furnace design and burner arrangement, but the high velocities of flame propagation accompanying rapid ignition are generally attained for a wide range of burning rates by supplying mixtures with the air-coal ratio varying from 1 to 3.5. Hence, the coal supplied by the blowing or pumping systems is usually stored in substation bins, from which coal is fed at the desired rate to a primary-air stream which leads to the furnace. Usually this feed consists of a feed screw with a variable-speed drive. Where many small furnaces are to be supplied, a single primary-air fan and feed screw may supply a third type of system, i.e., the air-mixture system.

Air-Mixture System. Fig. 3 shows that this system handles a mixture of air and coal in the ratio of 4 to 1, by weight, far different from the others. High velocity is necessary to keep the coal uniformly distributed. As portions of the stream are tapped off, the size of conduit is reduced to maintain sufficient velocity. A nonreturn system has been used (22), but the difficulty of balancing the rates of coal supply and consumption usually makes it necessary to supply an excess of mixture to a loop system. The coal and air which are not burned are fed back to the substation, either to be recirculated immediately or to be separated; in the latter case, the coal is stored in the substation bin and the air is vented to the atmosphere. An interlocking control between fan and feed screw is desirable to maintain a fixed air-coal ratio regardless of feed rate.

A recent installation of the air-mixture system (27) eliminates the usual bin and pulverized-coal feeders by supplying the coal to the system directly from the pulverizing mill. It is a modification of the Covert (28) system in that the unused coal is not returned to a bin but is continuously circulated until used. The fan which supplies air to the mill was installed with sufficient capacity to circulate the air-coal mixture around the loop in addition. This modification may introduce the same advantages of compactness and control which the direct-firing system brought to boiler-furnace practice when it replaced the bin-and-feeder system.

The air-mixture system has been involved in many of the installations where pulverized-coal explosions have occurred. In some cases the system itself has been shown to be definitely the cause of the ignition of an air-coal mixture. Fig. 3 shows that this system operates with a mixture which lies in the range of ignitible air-coal mixtures. However, the pressures which can be developed by explosion of these mixtures are not so high that reasonably sturdy construction cannot withstand them. Experience with explosions in pulverizing mills has shown that it is practicable to build them to withstand the most explosive of air-coal mixtures. Many of the early installations of the air-mixture system employed light riveted steel pipe which ruptured easily when slight explosions occurred. This allowed the mixture to escape and serious explosions of the unconfined mixture resulted.

Reasonable care in operation, in addition to sturdy construction, eliminates practically all danger from explosions. The known inflammability of oil and gas naturally encourages care in handling; similar care should be exercised with air-coal mixtures.

TABLE 2 COMPARISON OF COAL-DISTRIBUTION SYSTEMS[a]

System	Air-coal ratio, lb per lb	Velocity at prime mover,[b] fpm	Initial pressure, psi	Power consumption[c] hp-hr	Feeder power per furnace, hp
Blowing:					
Grindle.........	0.0265	1300–1500	10		
Quigley........	0.025–0.050	6450–9350	30–60	600–700[d]	0.5–1.0
Amsler-Morton[e]	0.026	2360	45–70		
Pumping:					
Pulco..........	.....	..	..	800–1000	0.5–1.0
Fuller-Kinyon...	0.024–0.038	800	30		
Air Mixture:					
Holbeck........	3.8–4.7	5000	0.75		
Covert.........	.....	5000	0.75	3000–3500	None required
Bergman.......	.....	5000	0.75		
Screw Conveyer...	.....	..	..	800–1000	0.5–1.0

[a] Based on figures given by Harvey (27), except where otherwise noted.
[b] At the end of the line the pressure is normally near atmospheric, and the resulting expansion of air may make the velocity several times this figure, depending upon the initial pressure.
[c] Based on 5–6 tons per hr; distance of 1000 ft; 20-hr day.
[d] The blowing systems are intermittent and operate approximately 20 per cent of the time; all others are continuous.
[e] Data for the Amsler-Morton system are based on figures given by Herington (24).

Comparison of Systems. No one system is universally applicable; considerations peculiar to each installation are the deciding factors. The characteristics of the systems are shown in Table 2.

In general, the higher the velocity, the higher the maintenance costs, because of the erosion at bends, valve parts, fans, pumps, etc. For this reason the simple screw conveyer is cheapest to operate. Most expensive to operate both on the power and maintenance basis is the air-mixture system. Blizard (23) quotes the maintenance cost of typical screw-conveyer and air-mixture systems as 15 cents and 86 cents per ton, respectively. In connection with the power requirement of the air-mixture system he says: "The ratio of power used to distribute the coal-and-air mixture per ton of coal used will increase with the distance that the mixture is transported and decrease with the load factor on the plant. This system is therefore better adapted for supplying a plant which requires a fairly constant supply for furnaces not too widely scattered." For one blowing system, Herington (24) reports "total cost of repair and maintenance of 3.8 cents per ton," and on another "total cost of transporting the coal as 1 cent per ton."

Existing Applications of Pulverized Coal

One of the most successful of the systems employing the air-mixture transport system was that installed at the Baldwin Locomotive Works in 1922, which at present supplies 22 tons of coal per day to 31 furnaces with hearth sizes ranging from 3 × 6 to 6 × 11 ft, handling forgings ranging from 20 to 100 lb each. The coal is dried and pulverized at a central preparation plant from which it is supplied to four substations, including one at the boiler plant, by a compressed-air blowing system with lines 300 to 500 ft long. A number of furnaces are supplied from each substation by means of fans and feed screws which feed a coal-air mixture through the loop of a circulating system.

Developments and improvements in this system have been described (19, 25). The furnaces are similar to that shown in Fig. 2. Primary air and coal are regulated by a manually operated valve in the branch line, and secondary air is induced by the combined action of the primary jet and a stack. Some stacks serve two or more adjacent furnaces. Compared to the latest forge furnaces, this plant leaves much to be desired in the way of automatic control, but considering its many years of service it has been a smooth-running economical system.

Two large forge furnaces using pulverized coal have been in operation at the Roanoke shops (26) of the Norfolk and Western Railway since 1932. Coal is pulverized at a central plant for boiler furnaces and for the forge furnaces. A compressed-air blowing system transports it to the furnace bins through 500 ft of 4-in. pipe. Screw feeders supply coal from bins to the air stream leading to the furnaces which are run in tandem, the flue gases from both passing to a single stack by way of a preheating furnace used in conjunction with the main furnaces.

Costs

With due consideration for preparation costs, pulverized coal is often the most economical fuel for metallurgical purposes. The margin between the unit delivered costs of raw coal and other fuels can be applied toward drying, grinding, and transporting the prepared coal to the furnaces and still leave savings. Fig. 4 shows the magnitude of the margin. For a given coal-cost ordinate, all points above this ordinate on the curve lie in a region of possible savings; the margin is readily seen for any given set of fuel prices. There is some minimum size of

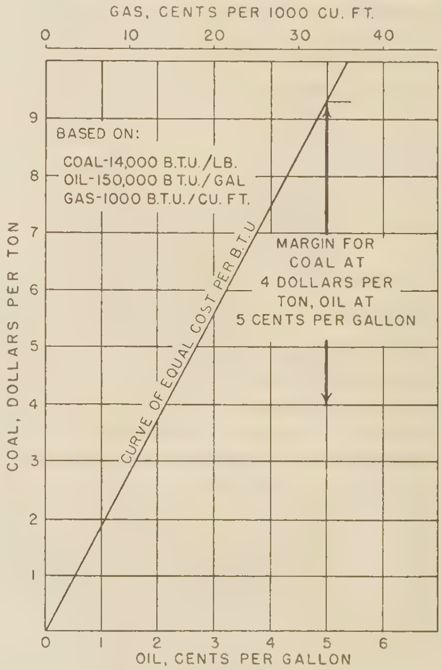


Fig. 4 Comparative Fuel Costs

installation for each set of conditions under which the margin would be insufficient to cover depreciation, power, and overhead costs, and thus investment in preparation and transporting equipment would be unsound. Even in some such cases savings could be made if properly prepared coal could be obtained from some local source already grinding for some other coal-consuming installation and possessing reserve grinding facilities.

Laboratory Study

Fig. 5 shows the typical small slot-type forge furnace used in the laboratory to study the application of pulverized coal in forge furnaces. This furnace was loaned through the courtesy of the Columbus Bolt Works and is one of a number of such furnaces now fired by oil at that plant. No alterations were made in the furnace other than the addition of a hood and stack connection and enlargement of the burner ports. The furnace hearth was $26^1/_2 \times 40$ in. and the volume of the combustion space was 15 cu ft.

The furnace was equipped with two burners located on opposite ends of the chamber in the same manner as when oil was used.

These burners were offset 3 in. horizontally in order to avoid direct impingement of the opposing flames as well as to induce a swirling flame within the chamber. Fig. 6 is a sectional view of a burner. A pipe carrying the primary-air-coal mixture discharged into the throat of the venturi-shaped burner port. The annular throat between the burner pipe and the port provided entry for a rapidly rotating envelope of secondary air; rotation was produced by supplying air to the wind box tangentially. Wind-box pressure was approximately 1 in. of water; furnace pressure ranged from 0.2 to 0.4 in. of water.

Usually, for convenience, a gas flame was used for starting. The gas pilot consisting of a $^1/_8$-in-diam orifice was incorporated into the tee in the coal supply line and was connected to the gas

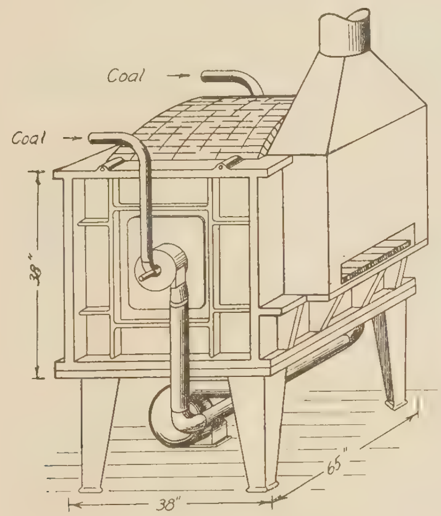


Fig. 5 Slot-Type Forge Furnace Tested in Laboratory

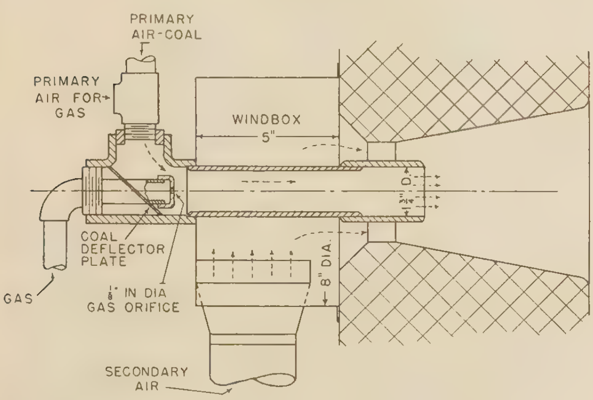


Fig. 6 Experimental Pulverized-Coal Burner Used on Laboratory Forge Furnace

main which supplies natural gas at a pressure of 10 psi. When it was desired to use gas for preheating, the pilot could be operated as an inspirating premixing burner which could obtain air through an open branch of a tee in the coal supply line just above the burner. When coal was fired this branch was closed.

The furnace has been started also by building a small fire of wood or coal on the hearth and firing the pulverized coal over the incandescent bed thus formed. When using the gas pilot for starting, the gas could be turned off after the adjacent refractory reached 1500 F, since ignition of the coal was then self-sustaining.

Preliminary trials were made with a simple atmospheric-type burner consisting of a central coal jet discharging into the venturi-shaped port. This arrangement was found inadequate to inspirate the necessary secondary air. If the momentum of the primary-air-coal jet had been made high enough sufficient air could have been entrained; but then the burner would have been unsatisfactory because of the excessive velocity of the coal particles which would have accompanied the air velocity necessary to develop sufficient momentum. Hence the forced-air burner in Fig. 6 was designed consisting of the wind box at each burner to which secondary air was supplied by a separate fan located underneath the furnace, as shown in Fig. 5. This provided positive, controlled, and measurable secondary air in any quantity desired.

Mixing of the primary-air-coal and the secondary-air streams was attempted by means of deflecting vanes located inside the tip of the coal supply pipe. It was found that if the vanes were located too near the tip the many small streams of coal into which the main coal stream was divided were caused to impinge on the hot refractory port wall; this resulted in an accumulation of coke. If the vanes were moved farther back from the tip, the mixing effect was confined to the primary-air stream itself, and there was little extra mixing with the secondary air.

A further attempt at mixing was made by installing a $^1/_2$-in. pipe concentric with the primary-air-coal pipe. A part of the secondary air was diverted into this pipe in order to introduce air at the center as well as at the periphery of the central coal jet. Ignition with this arrangement was not so rapid as with the simple straight pipe shown in Fig. 6.

The best burner performance of the modifications tested was the simple straight pipe burner shown in Fig. 6. Improved performance may be obtained from additional study of this problem.

A disk-type feeder supplied pulverized coal from a bin to the primary-air fan which delivered the resulting mixture through standard pipe to the burners. The feed rate was controlled by means of a thumbscrew adjustment on the feeder and the primary air was regulated by a gate valve in the blower-discharge pipe. Secondary air was supplied by a separate blower, control of which was obtained by a butterfly valve at the blower inlet. All air quantities were measured by means of plate orifices and liquid manometers.

In early tests, the coal and primary air were supplied by an experimental aeration-type feeder, but irregularities in coal feed made combustion irregular and calculations uncertain; hence pending further development on this feeder it was replaced by the disk feeder.

Furnace temperature $1^1/_2$ in. above the hearth was measured by means of a Pt, Pt-Rh thermocouple in a refractory protecting tube which was inserted through the back wall to project 7 in. inside the chamber. A similar protected couple with water-cooled leads was inserted through the slot to make temperature traverses on the hearth. By means of an optical pyrometer, frequent check readings were made; this instrument was frequently checked against a laboratory standard.

A hood installed at the top of the usual furnace-exhaust opening was connected to the laboratory breeching through 20 ft of 8-in. pipe. An available draft was thus obtained of from 0.01 to 0.04 in. of water. The usual air screen was provided for deflecting the flame issuing from the slot.

An attempt was made to determine the effect of fineness and volatile matter of the coal and of the amount of excess air on the combustion efficiency. In order to do this, the amount of unburned carbon leaving the furnace was used as a criterion of the combustion efficiency. Calculation for this quantity was based on the average flue-gas analysis, weights of air and coal supplied, and the carbon content of the coal fired. The carbon loss is given by:

$$\text{Loss, per cent of carbon fired} = 100 \times \left[1 - \frac{(\text{Mols carbon, flue gas})/\text{Mols nitrogen, flue gas})}{(\text{Mols carbon, fired})/(\text{Mols nitrogen, supplied})}\right]$$

$$\text{or} \quad \text{Loss, per cent of carbon fired} = 100 \times \left[1 - \frac{(CO_2/N_2)}{\dfrac{WC/12}{(0.769\,A)/28}}\right]$$

where CO_2 = carbon dioxide, dry flue gas, per cent by volume
N_2 = nitrogen, dry flue gas, per cent by volume
W = coal-feed rate, lb per hr
C = carbon content of coal, lb per lb
A = air-supply rate, lb per hr

The method cannot be considered accurate. It neglects the nitrogen and sulphur in the fuel, but for coals used, this error was less than 0.2 per cent. Also, a maximum error in the value for percentage carbon loss of ± 50 per cent is possible owing to small experimental errors in the five measured vairables. However, direct measurement of the unburned carbon in the flue gas was not attempted because of the known difficulties which are often encountered in securing accurate samples Instead, an average calculated loss of unburned carbon was based on a number of tests for each set of conditions.

The logical point at which to determine the average flue-gas analysis is at the slot, as any burning which occurs after the gases leave the slot does no useful heating. However, attempts to determine the average flue-gas composition from careful traverses of the gases at the slot by means of a water-cooled sampling tube were unsuccessful. Fluctuations of velocity and gas composition caused the calculations based on average analyses from these traverses to be anomalous. Hence, it was necessary to sample the gases at the top of the hood after mixing of the stratified gases was obtained. It was recognized that this included the products of combustion from some useless burning which occurred after the gases left the slot, but for comparative tests on the important variables this was deemed acceptable. During the combustion tests the work opening at the base of the hood was closed off by a refractory-lined door and the air screen was shut off to avoid dilution of the gases before sampling.

The primary-air-coal ratio employed for most tests was approximately 2. Variation of this ratio from 1.6 to 2.5 showed no consistent change in the amount of unburned carbon in the flue gases. At a coal-feed rate of 50 lb per hr and a primary-air-coal ratio of 2, the velocity of the primary-air stream in each burner was 663 fpm. The velocity of secondary air at the burner port throat varied from 1430 to 1880 fpm.

Effect of Type of Coal

Table 3 shows the proximate analyses of the coals investigated. Tests were run with the high-volatile Elkhorn coal pulverized to 93, 85, and 72 per cent minus 200 mesh and with the low-volatile coal pulverized to 93 per cent minus 200 mesh. Observations

TABLE 3 PROXIMATE ANALYSES OF COALS TESTED
(Basis: as fired)

	Elkhorn	Pocahontas No. 3
Moisture, per cent	0.2	0.1
Volatile matter, per cent	35.2	20.8
Fixed carbon, per cent	61.0	74.6
Ash, per cent	3.6	4.5
Sulphur, per cent	0.8	0.6
Btu per pound	14850	14900
Ash-softening temperature,[a] F	2510	2300
Hardgrove grindability[b]	46–53	101–111
Percentage through 200 mesh	93	93
Specific surface,[c] sq cm per gm	2870	5280

[a] Average for mine as reported by Bureau of Mines.
[b] From "Grindability of Coals," Bulletin 3-241, The Babcock and Wilcox Company, New York, N. Y., 1938.
[c] Determined by Lea-Nurse air-permeability method.

and results of these tests showed that, when firing the coal pulverized to 93 per cent through 200 mesh, ignition took place close to the burner tip and resulted in a short bushy flame. With the coarser coal ignition occurred farther from the burner tip and the flame was longer and more luminous. Because for a given air quantity the coal particles remained in the furnace for the same length of time regardless of fineness, the increase in flame luminosity was an indication of higher loss of carbon.

Below approximately 25 per cent excess air, difficulty was encountered with coking in the burner port because the secondary-air velocity was not sufficient to sweep separating coal particles into the furnace before they impinged on the hot refractory wall of the port. Above 50 per cent excess air, the loss because of unburned carbon was not decreased and the furnace temperature could not be maintained. Between 25 and 50 per cent excess air no consistent relation was observed between excess air and unburned carbon. Hence for the particular burners and furnace, excess air should be held to the lowest value which will not introduce coking. It is possible that the burner port could be so designed that coking would not occur. Excess air could then be carried at an even lower value than was possible for the laboratory burners and the limiting minimum excess air would then be that at which loss due to unburned carbon began to increase.

The average flue-gas analyses and losses of unburned carbon were as follows:

Coal	Fineness, through 200 mesh, per cent	Carbon loss of coal fired, per cent	Gas composition, by volume, per cent CO_2	O_2	CO	Excess air, per cent
Elkhorn	93	3	13.2	5.6	0.0	36
Elkhorn	85	5	13.0	6.0	0.0	38
Elkhorn	72	6	12.9	6.1	0.0	40
Pocahontas No. 3	93	6	13.4	5.7	0.0	35

The comparatively small loss of carbon for the coarser high-volatile coal and for the low-volatile coal may be owing to the presence of the incandescent refractories on the hearth and at the slot. Unburned carbon particles which fail to burn in suspension will have increased opportunity for burning if they deposit at these points.

The lower rate of flame propagation of the low-volatile coal tends to increase the loss of unburned carbon. However, this tendency was partly offset by the large amount of surface which this coal had as a result of its high grindability. Table 3 shows that the values of specific surface, square centimeters per gram, for the high- and low-volatile coals when pulverized to 93 per cent through 200 mesh were 2870 and 5280, respectively. The great surface which the low-volatile coal presented for reaction with oxygen made for rapid ignition and rapid burning.

When firing the high-volatile coal pulverized to 93 per cent through 200 mesh at a rate of 50 lb per hr, the furnace could be heated from room temperature to 2300 F in $1^1/_4$ hr. After the refractory was heated to thermal equilibrium, the furnace could be maintained at 2350 F when empty by firing at the rate of 32 lb per hr. With the average furnace temperature at 2400 F, a load of twenty 1-in. round steel bars 3 ft long projecting into the furnace 18 in., was heated to 2200 F in 10 min as indicated by the optical pyrometer. The coal-feed rate was 45 lb per hr. However, the furnace had not yet reached thermal equilibrium when this test was made.

Examination of the bars after removal at the end of the 10-min showed that scaling was light and that the thin deposit of ash on the bars fell off with the scale during handling. The coal-consumption rate for this test was approximately 170 lb per ton of steel heated. Consumption in production would be higher than this value because of the fuel consumed during stand-by and heating up periods.

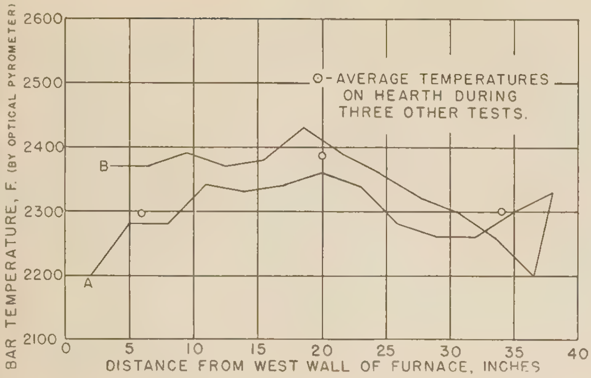


Fig. 7 Temperature Variation of 1-In. Bars on Hearth
(Temperatures measured as alternate bars were removed, beginning at *A* and ending at *B*.)

Temperature Variation on Hearth

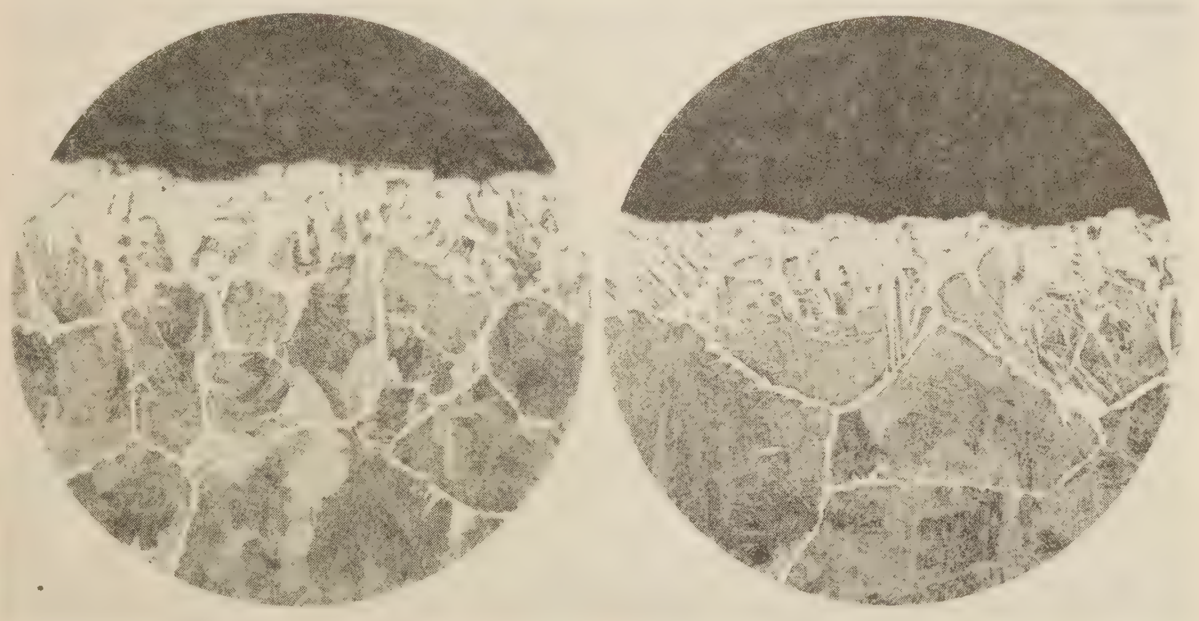
Temperature traverses on the hearth showed the center to be higher by 50 to 100 deg than the sides. This was caused by too vigorous burning of the opposing flames as they impinged above the center of the hearth. Greater horizontal distance between the burner center lines than the 3 in. used for these tests would alleviate this condition, although some difficulty might then be encountered because of impingement of the particles on the refractory.

Fig. 7 shows the results of a load test which was made in order to show the effect of this temperature variation on heated stock. Twenty-five 1-in. round steel bars were heated in the furnace for 15 min and then removed in order; the temperature of each bar was read within 5 sec after removal by means of an optical pyrometer. Care was taken to scrape off a section of scale to permit sighting the true surface of the bar. Close agreement between the curves for alternate removal from left to right and then back, right to left, indicates consistent variation of the temperature of the bars on the hearth. The plotted points in Fig. 7, show the average hearth temperature as measured by means of the optical pyrometer at intervals during three other tests.

Decarburization

In the section describing the effect of atmospheres, scaling was said to be a lesser evil than decarburization. Whereas scaling entails loss of material and requires extra labor for its removal, decarburization leaves a soft surface on the stock because of diffusion of the carbon from the steel into the furnace atmosphere. This damaged material must sometimes be removed by machining, which is expensive and likewise entails loss of material. If not removed, it may develop fatigue cracks in service which may propagate themselves into the unaffected structure of the steel. Hence decarburization should be eliminated if possible or at least kept to a minimum.

Fig. 8 shows the extent of decarburization on the surface of two samples from the same bar of S.A.E. 1045 steel heated in the laboratory furnace and in an identical oil-fired furnace. The oil-fired furnace was operating under actual production conditions and was being operated at a rather high temperature. For this reason, the sample heated in the oil-fired furnace had to be removed after 5 min to avoid danger of burning. The white area near the surface from which most of the carbon has diffused is of approximately the same extent for both fuels. The depth of decarburization is about 0.005 in. in both. Hence the steel heated by pulverized coal suffered no more from decarburization than that heated with oil.



Pulverized-Coal Furnace
Furnace temperature, 2300 F
Sample in furnace, 8 min
Temperature of sample when removed, 2200 F

Oil Furnace
Furnace temperature, 2530 F
Sample in furnace, 5 min
Temperature of sample when removed, 2200 F

Fig. 8 Photomicrographs Showing Surface Decarburization on S.A.E. 1045 Steel Heated by Pulverized-Coal and Oil-Fired Furnaces; × 100. Picric-Acid Etch
(Temperatures measured with optical pyrometer.)

Effect of Piping

Irregular feeding of the coal was encountered with both feeders which were used. Minute irregularities were not objectionable, but, periodically, dense masses of coal were supplied to the furnace at such a rate that excessive smoke and flame were blown out of the slot. The arrangement of piping was found to be responsible for this difficulty.

Originally the coal was supplied through standard $1^1/_2$-in. pipe with a total length of 22 ft to each burner including two long-radius bends and 12 ft of horizontal run. For a feed rate of 25 lb per hr to each burner and an air-coal ratio of 2, the velocity in each pipe was 850 fpm. This velocity was too low for proper transport and some of the coal particles dropped to the bottom of the pipe; usually the bottom quarter of the pipe was filled with coal.

At irregular intervals portions of this fluffy mass of coal were picked up by the air stream and fed to the furnace at a rate in excess of the possible burning rate. This condition was aggravated when any coal was used which had a tendency to adhere to the top and sides of the pipe owing to its inherent stickiness or possibly to oil treatment of the coal. Then the coal would adhere in the vertical as well as the horizontal pipes, especially on small projections such as at joints or in the regulating valves. When portions of such coal masses were dislodged the smoking and flaming from the slot was severe.

The difficulty caused by settling in the piping was practically eliminated by replacing the $1^1/_2$-in. pipe with $^3/_4$-in. pipe and reducing the length of horizontal pipe from 12 to 3 ft by relocating the feeder. Even with the resulting velocity of 3100 fpm, the sticking coals adhered to the walls of the pipes and increased the resistance of the system.

It is probable that with sufficient fan capacity the velocity in the long horizontal pipes could have been maintained at such a point that all tendency of the coal to separate would have been eliminated. However, the power consumption would have been high. Minimizing the length of horizontal pipe is the more logical and economical solution.

Inherent stickiness is not uncommon with high-grindability coals which normally produce a large amount of superfines when ground. The property is most troublesome where the fines are given an opportunity to pack as in storage bins and feeders. The agglomerated mass then fails to flow uniformly or may fail to feed entirely. Evidence that oil treatment increased stickiness was not conclusive in the laboratory tests. Oil treatment to the amount of 1 gal per ton seems small considering the enormous surface presented by pulverized coal. However, sieve tests for fineness were definitely more difficult with oil-treated medium-grindability coal than with untreated coal. With the pulverizing plant so adjusted that it was grinding a medium-grindability high-volatile coal to 93 per cent through 200 mesh, a lot of the same coal was ground which had been oil-treated for domestic-stoker use. Sieve tests on a sample collected during grinding gave only 60 per cent through 200 mesh. The residue on the 200 mesh screen was caked; the agglomerated masses were composed of many fine particles which should have passed through the 200-mesh screen. A similar experience in sieving a sample of a low-volatile, high-grindability coal was surmounted by wet-sieving the residue on the 200-mesh screen with petroleum ether. The fineness as determined by wet-sieving was in the range normally obtained with the pulverizer settings which were used in grinding this sample. Prolonged mechanical sieving with such sticky samples was found to give nearly the same fineness as the wet-sieving method; this indicates that the agglomerated masses of fines can be broken down mechanically if they are subjected to agitation.

Summary

Pulverized coal has demonstrated its suitability on many types and sizes of forging furnaces. For most sections of the country pulverized coal is the most economical fuel. It has the same advantage of fluidity as gas and oil which permits easy clean handling in pipes and ready mixing with air. It gives a flame of high emissivity which provides maximum heat transfer by radiation.

Regardless of its economy the suitability of coal for forging makes it of great current importance when the supply of oil and gas is so limited. Also, conservation of limited supplies of oil and natural gas to the uses for which they alone are suited is recognized as sound even in times of peace. By continuing and accelerating progress in the utilization of pulverized coal in forging and other metallurgical industries, the coal producers and equipment and furnace manufacturers can make a substantial contribution to this trend toward conservation.

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Discussion

R. J. Bender.[4] The writer has had the good fortune of working for some of the leading pulverized-coal-equipment manufacturers and has constantly wondered why there seemed to be such a marked barrier between the departments that were designing equipment for power-plant use and those that devoted their efforts to metallurgical work. For boiler firing, it seems to be quite well established that active turbulence is one of the essential rules for proper combustion of pulverized coal; it is difficult to admit that the same rule will not apply to metallurgical furnaces, regardless of the restricted width or the short length of the combustion space. It may sound paradoxical, but it is believed that the design of pulverized-coal burners is still in its infancy; much is still to be accomplished in the way of controlling turbulence and in flame placement.

The authors should be congratulated for having prepared a worthy contribution to an up-to-date matter of study, in line with the usual high standards of the work done by Battelle Institute.

A. J. Grindle.[5] It has been the writer's experience in metallurgical furnaces that considerably less oxidation or scaling of metal results from the use of pulverized coal as fuel than with oil or gas. It is believed this is due to the fact that less excess air is used than is required with oil or gas.

It has also been the experience of the writer, that with small furnaces, the coal must be pulverized finer than is necessary for large combustion chambers; and with the proper feeding and burning equipment, a soft, penetrating heat can be maintained.

The statement in the paper that equipment manufacturers and the coal industry have not "continued efforts toward development and improvement of coal-firing equipment" is true, and the co-operation of these interests would result in increased use of coal and the sale of equipment.

It has been found that velocities of the coal-carrying air, to prevent flame pulsation, should be between 3500 fpm for finer pulverized coal, or for short runs, and 4500 fpm for coal not so fine, or long runs. This velocity should be reduced at the burner nozzle to around 1000 fpm or less, if the furnace design permits, without coking. Ordinarily a burner nozzle of the same rectangular shape as the cross section of the furnace tends to fill the furnace better. For metallurgical work, a burner, so designed that all of the air and pulverized coal are mixed before entering the furnace, produces a more neutral or less oxidizing flame and requires less excess air.

[4] Fuel Oil Engineer, Sinclair Refining Company, Chicago, Ill. Jun. A.S.M.E.

[5] Pulverized Fuel Engineer, Whiting Corporation, Harvey, Ill.

It is important with pulverized-coal firing that the products of combustion be vented to dust collectors or outside of the building, and no application of pulverized coal should be made unless this is done.

Although the air mixture or circulating system might have advantages in certain ways and for certain applications, the writer would recommend, first, a unit pulverizing system in which the raw coal is conveyed to storage hoppers over individual pulverizers by a mechanical or pneumatic conveying system, or, as a second choice, a storage of pulverized coal at each furnace, supplied from a central pulverizing plant. The circulating system has one inherent disadvantage in that, when a plant is running at low capacities, the complete system must be operated with sufficient coal and air flowing through the pipes to give the proper mixture, even though only one furnace of a large group might be operating.

The paper states that "pulverized coal, when mixed with air, can be made to flow through standard pipe over distances up to 1500 ft." It might be interesting to note that installations are operating without difficulty where the pipe line is as much as 5000 ft long. In all of the pneumatic conveying systems where coal is conveyed with small volumes of air, as compared to the weight of coal, the flow is in alternate slugs of air and coal and not in a uniform mixture, even though the coal may leave the feeding or injection point uniformly.

The writer's experience has been that the maintenance costs for the various conveying systems are less with the blowing or pump system than with screw conveyers, and the air-mixture or circulating system has the highest costs, both for power and maintenance.

The writer believes that small unit pulverizers with capacities as low as 50 lb per hr or probably less can be developed, and a pneumatic conveying system used for supplying the raw coal to the pulverizer storage, thus providing a pulverized-coal-firing system for small furnaces equal in operating results to the present installations of large pulverizers on melting and heat-treating furnaces. This would be a much more flexible system than the circulating system and probably less expensive in first cost as well as cost of operation and maintenance.

Another possibility, if the coal industry were willing to co-operate, would be to pulverize coal at the mine or central points and deliver to the consumer by tank car or tank truck, both of which units have been perfected. If a supply of pulverized coal were available, it could be delivered to laundries, small industrial plants, forge shops, and many other small users of fuel who would then be willing to use pulverized coal rather than oil or gas, because the firing equipment could then be sold on a competitive basis with oil-burning equipment, and the fuel costs would be considerably lower.

It is the writer's opinion that properly designed burners will prevent coking without using an excessive amount of air. In fact, in high-temperature furnaces where metal is melted, we at times use less than the theoretical amount of air required and do not have coking trouble. It is very important that a steady flow of coal be maintained, and this can be done with any of the three systems described in the paper, by the proper feeding mechanisms and air velocities in the piping. Pipe sizes must be provided as large as possible to prevent excessive pressure losses, but small enough to maintain the necessary velocities to keep the pipe clean without supplying an excess of air for combustion.

It is apparent from past experience in the use of pulverized coal for forging furnaces and in view of improvements made in pulverized-fuel equipment and knowledge gained by engineers working in this field, that pulverized coal is a suitable fuel for many types and sizes of forging furnaces. The authors of this

paper should be complimented on their efforts and encouraged to continue their research and experimental work.

C. F. Herington.[6] The need for unreserved efficiency and conservation of resources in our industrial plants has never been so urgent as during the past year. With the constant, increasing demand for production, there exists a pronounced deficiency in practically all supplies required to meet this demand. This is true in the case of fuels, and it is frequently necessary to curtail the quantity delivered to our industries in order that domestic needs may be met.

The savings secured by the substitution of pulverized coal for other fuels have been most encouraging and this discussion should prove to be most timely.

Furnace Design. In the early days of pulverized-coal installations, it was imperative that the question of making furnace changes be carefully avoided; just a statement was made to the effect that we simply change the burners. In a preponderance of installations, furnace design could not be improved upon: "Didn't his father and grandfather use it and make a living?" In a few cases, changing the burner did show a saving in fuel, refractories, and heating time; because a good pulverized-coal burner proved to be a good burner for natural gas and fuel oil.

However, we are convinced that the most economical way to install pulverized coal on any furnace is to design the furnace correctly for the kind of heating desired and to fire it in such a manner and with the correct elements to achieve 100 per cent perfect combustion.

The authors make reference to the use of a low velocity for admission of the coal to the furnace. In the early days of pulverized-coal installations, many means were tried to achieve a low velocity; one burner design moved the particles of coal dust back and forth horizontally before it became dizzy enough to enter the furnace, and so today we are using higher velocities and getting results.

Most of the users of pulverized coal believe the success of this method of firing lay in the design of the burner, and according to the patent-office files, most of the inventors did not stop at one design but made many. The greatest number of burner designs seemed to consist of all kinds of gadgets to be inserted in the burner to slow up the velocity, to mix, change direction, everything but to leave the mixture alone.

The writer has found that a simple piece of pipe hammered flat at the end proved to be a very successful burner, the secondary-air pipe being treated in like manner and entering the furnace directly under the coal pipe.

The following are the essentials to achieve perfect combustion in a pulverized-coal-fired furnace:

(*a*) Fineness of coal for the purpose.
(*b*) Dry coal.
(*c*) Constant and uniform feeding of the coal into burner.
(*d*) Burner design.
(*e*) Separately controlled secondary air.
(*f*) Furnace design.
(*g*) Larger waste-gas flues.
(*h*) Constant volume and pressure of primary air and coal.

There is a distinct relationship between the successful burning of pulverized coal in any furnace, and one in which the conditions are met with reference to furnace design, adequate waste-gas flue areas, etc.

Following is an outline of a few installations covering the essentials as heretofore stated:

Furnace No. 1, 14 ft wide × 45 ft long, is used for melting copper from blister to pour into anodes. First, three circular burners were tried with the coal and primary air entering the center of a 10-in-diam pipe which conveyed the secondary air, the idea being that the primary air and coal would be completely surrounded by the secondary air as it entered the furnace. Actually, the upper half of the secondary air contained in the 10-in. pipe rose toward the furnace roof as it left the burner mouth away from the coal dust and the coal immediately started to drop toward the furnace hearth, meeting the lower one half of the secondary air, which was not enough to support combustion. As a result ash and carbon particles dropped on the bath.

The ratios obtained were as follows:

With the three circular burners, it required 250 lb of coal per ton of copper melted.

By changing the burner to a flat fishtail type, also the secondary air, and using only one burner, it required 160 lb of coal per ton of copper melted.

Furnace No. 2 was a furnace for reheating rails, Fig. 9 of this discussion. When we first saw the installation, two beater mills were being used, grinding coal to 70 per cent through a 200-mesh screen, each mill delivering the coal into two branches, consisting of round pipes aimed directly at a bridge wall at a point where the heated rail was ready to be pushed out for rolling. We redesigned the entire furnace, from the foundations, leaving only the entrance and discharge doors. We shortened the furnace 6 ft from the front wall, to the bridge wall; raised the flat roof so the burners entered at 7 ft above the floor. The roof was 5 ft above the rails, then sloped abruptly toward the pusher end of the furnace so that, for one third of the distance back, the roof was only 12 in. above the rails.

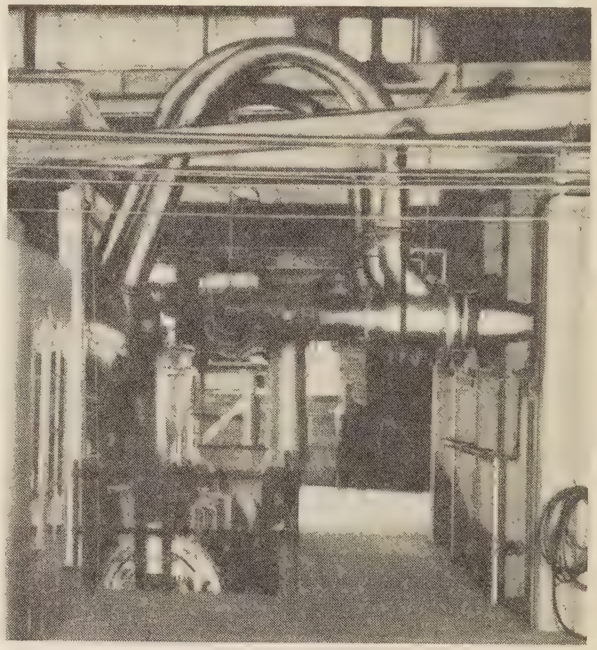

Fig. 9 Unit Pulverizer System Applied to Continuous Rail-Heating Furnace Showing Burners and Piping

A new bowl pulverizer was installed, using the one exhauster to branch into four burners. These burners were designed to have a velocity of 6000 fpm, each burning 21 lb of coal per min. The waste-gas flues were enlarged and surrounded with air-cooled wall tile. This furnace operates from 7:00 a.m. to 3:30 p.m., and starting the furnace at 7:00 a.m. the first rail is ready to roll at 7:30 a.m.

[6] The Amsler Morton Company, Pittsburgh, Pa.

Automatic temperature and furnace-pressure controls are used on this furnace to eliminate sticking of rails.

At the end of one year, the management reported no expenditures for maintenance either on the furnace or pulverized-coal equipment, and the net result showed an increase in tonnage of 15 to 20 per cent with a saving of 38 per cent in fuel. The complete change of demolishing the old furnace and placing a new furnace in operation required 21 days.

Furnace No. 3 was a new forge furnace which we built, with a hearth 8 × 16 ft of the two-door batch type, used for heating 8-in. × 8-in. × 7-ft alloy-steel blooms from 60 to 2250 F. These furnaces are also equipped with automatic temperature and furnace-pressure controls.

After two months of operation the waste-gas flues opposite the burner filled up with ashes, requiring a shutdown. After a sample coal analysis had been made, we found that the fineness of the coal being used was 49$^1/_2$ per cent through a 200-mesh screen. The fuel requirements of the works at the present time leave a short period of 16 hr every week for a shutdown so that if pulverizers were turned up for finer grinding, the capacity in tons would not be sufficient to supply all of the furnaces. These pulverizers are 22 years old and grind about 9 tons of coal per hour. We are going to add a roller mill of 12-ton per hr capacity to the coal plant. This mill will provide 85 per cent through a 200-mesh screen, using a steam aerofin heater to dry the coal in the mill. This will allow the rotary coal drier and pulverizers to be used as spare.

Example No. 4 is a plant which has about 70 forge furnaces varying from a small slot to one as large as the one just described, besides walking-beam, pressed-steel, rivet-making, and other furnaces. These burn pulverized coal and were installed in 1918.

Here again, the fineness of the coal is a detriment to good forge-furnace work. Two pulverizers similar to that just described are installed, only their fineness is 70 to 73 per cent through 200 mesh. In this case the boiler plant requires 15 hr of pulverizing out of every 24 hr to furnish the correct amount of coal, leaving only 9 hr supply per day for the furnaces. With the setup of equipment at this plant, it would be impossible to change the fineness of the coal for the furnaces to 85 per cent. Since the 73 per cent is correct for the boilers, the furnace tonnage suffers by sparks chasing each other through the furnace and thereby lowering the furnace efficiency.

W. W. Pettibone.[7] Twenty years or more ago pulverized coal was used by a large number of iron and steel plants for many operations ranging from open-hearth furnaces, soaking pits, puddling furnaces, charge-and-draw and continuous-heating furnaces, sheet-and-pair furnaces, annealing furnaces, to large and small furnaces for forging operations, etc. Since then low-priced fuel oil and gas have replaced pulverized coal to a great extent with the possible exception of the malleable-iron or air furnace. The present situation with fuel oil and gas makes consideration of pulverized coal of interest to many whose experience does not extend back to the period when it was more universally used.

The continued use of it in some sheet-and-pair furnaces (such as at the Newport Rolling Mills, for the past 22 years or more) should be evidence that the fear of the effect of coal ash on steel heated with this fuel is greatly exaggerated. Proper selection of the coal will avoid detrimental fusion of the ash to the steel and what does deposit tends to act as a protection to the steel against excessive scaling or oxidation. Control of the furnace conditions to maintain a neutral or slightly reducing flame is not difficult with pulverized coal.

[7] Manager, Pulverizer Division, The Sims Company, Erie, Pa.

The writer would like to inquire whether the authors considered the use of a compressed-air siphon-type feeder in their experiments on the small forge furnace? These feeders were used on a number of furnaces of about this size at a plant he was connected with some years back. They gave very good control and uniform feeding, although the coal was all fed through a $^1/_2$-in. pipe, with the compressed air from a small nozzle serving to control the coal feed from a hopper and also as the primary air.

The writer would also like to suggest that consideration be given to preheating the secondary air by carrying the supply pipe through the hood over the front of the furnace. Any temperature gain obtained in that way will materially benefit the furnace operation, the steel, the combustion and will aid in maintaining a higher CO_2 in the furnace. A CO_2 content of 15 to 16 per cent was easily maintained with the compressed-air feeder and preheated secondary air on the small furnaces previously referred to, which were used for heating small bars for track-spike forging machines.

Where a large number of furnaces are involved, the proper pulverized-coal system to consider is the bin or storage system with a central plant supplying all furnaces. However, we have small unit pulverizers available today with capacities of 100 lb of coal per hr or less and from this minimum upward in generally convenient sizes. Where only a few furnaces are involved, these unit pulverizers offer a means not only of meeting the present fuel-oil and gas shortage but also of reducing the heating costs in many cases with equal furnace capacity and quality of product.

W. E. Reaser.[8] The writer wishes to inquire if, in the investigation which prompted this excellent paper, attention was directed toward the problem of control of the air-fuel ratio. No matter how large or how small the quantity of fuel fired into a combustion chamber, continuous precise proportioning by weight of the combustible with oxygen is as important as the design of burner, furnace, fineness of pulverization, and the various other factors which must be considered.

When burning pulverized coal, this problem deserves even greater consideration than when oil or gas is the fuel, because of the lack of an adequate device to control positively the feed of the pulverized material. Actually, modern combustion practice will demand "micrometer adjustment" of the air-fuel mixture, if over-all most economical operation is to be secured. To accomplish this goal, a continuous measure of excess air, best determined by analyzing the combustion gases for the O_2 content, is necessary. It should be emphasized that instruments are now available to perform this function.

If the research work on the forge furnaces did not include an examination of this factor it might be interesting to consider in future investigations of the subject.

W. C. Rehfuss.[9] This paper goes quite thoroughly into the matter of scaling, decarburization, dust, etc., with regard to their effect on the product, and also mentions other factors of vital interest. From a practical viewpoint, some of the considerations that impelled the writer and his associates about the year 1927 to undertake revamping a forge shop as large as that of the Baldwin Locomotive Works were forging costs, heating capacity of furnaces, furnace maintenance, safety and health conditions.

The cost of producing anything is usually the first consideration and, where it is possible to bring down that figure, the approach to a problem is indicated. When our job of experiment and conversion was completed, we were highly gratified to find that

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forging costs had dropped to the lowest point in the history of the Baldwin plant and have continued on that level up to the present time, covering a period of approximately 15 years.

When we started on our project, the shop was equipped with the Holbeck system of pulverized-coal production and distribution and a number of pulverized-coal furnaces of the single-door type with hearths averaging about 3 × 4 ft. The coal-control valves were a seat-and-disk type that did not prove satisfactory because of leaking and sticking; the burners had branch connections, one for the coal and primary-air mixture and the other for secondary air which came from blowers.

The furnaces had a slag slot in the rear and a door in front. All products of combustion, etc. made their exit at these two points. Although a hood was provided over the door, 90 per cent of the dust and gases went right past it and discharged into the shop proper. Furnace doors melted off in a week's time and the 18-in. wall opposite the burner was also pierced in that time.

Every time that a heavier slug of coal came through the coal loop, puffs of varying intensity occurred in every furnace on the loop concerned. With no other place to go these shot out under the doors and through the slag-slot openings. If the door happened to be open and the operator standing in front of it, he received the full benefit, often in the face. Accidents of this type occurred almost daily.

The new-type furnace which is described in the paper eliminated practically all of these faults, because of the fact that the furnace pressure was reduced to practically atmospheric by maintaining a simple draft control in the flue of the stack. As a result the products of combustion no longer went out through the door opening and, when the door was opened, cold atmospheric air did not rush in and chill the furnace; and puffs went on their way through to the stack instead of into the operator's face.

The low velocity of gases in the furnace was highly beneficial in that the chilling effect of excess air was kept to a minimum and sufficient time was permitted for the heat to be absorbed by the furnace walls and forgings being heated. The size and shape of the furnace was so designed that there was no impingement on the side walls, end wall opposite the burner, hearth, or arch. A glimpse at Fig. 2 in the paper will readily illustrate this point. One of the great objections to the use of blowers is the high velocity given to the products of combustion, which makes it difficult to avoid impingement in a small furnace. All secondary-air blowers, six of them, each driven by a 50-hp motor, were removed from the shop.

On the type of furnace which we designed, we utilized the energy of the primary air which was at a pressure of 7 oz to $^3/_4$ lb to induce the secondary air from the atmosphere, using a bell-mouth throat piece in order to procure maximum effect. This was assisted by the draft from the stack. The energy required for this purpose naturally slowed down the primary air so that we were not bothered by impingement from that source. In all cases the velocities were maintained at a point sufficient to keep the coal in suspension. That we were successful in doing this was evidenced by the fact that very little trouble was experienced with coking around the burner.

At the time we started on the change-over there were numerous oil-burning furnaces also in the shop. These were all replaced with pulverized-coal furnaces. This involved a design of perhaps half a dozen types, all of which fundamentally followed the design of that shown in Fig. 2. All oil, gas, and blast lines were eliminated from the shops entirely, and lighting up was accomplished by means of a small wood fire built on the hearth in front of the burner.

The heating capacity of the new furnaces was heavily increased and the time required to heat material decidedly reduced so that the production capacity of the working forces per man-hour went to a higher point.

An interesting observation was in the production of steel-bushing forgings, approximately 14 to 16 in. OD and 8 to 10 in. ID × 10 to 12 in. long. When the solid blanks from which they were made were heated in an oil-fired furnace, the forgings in all cases looked much hotter when placed under the press. However, when the slug was pushed out through the die, it could be seen that the interior of the forging was very much cooler than was the case when the same type of piece was heated in the pulverized-coal furnace. This bears out the comments given in the paper on that point.

It was generally noted throughout the forge shop that in all cases forgings heated with pulverized coal could be worked longer under the hammer than those heated with oil because of the fact that the heat had penetrated well to the center. This resulted in less reheats, and, in the case of simple forgings, eliminated reheats entirely. More and better forgings naturally followed.

With the elimination of darting puffs of hot gases and flame through the doors of the furnaces, and the removal of the discharge of gases and ash into the shop, the operator could perform his duties with much greater comfort and a far healthier atmosphere in which to work.

Many expressions of satisfaction came from the operators in the shop, men who were used to gases, dust, heat, hard work, and frequent burns. Due to the great improvement in working conditions their lot was decidedly better and when new help was needed there no longer was any difficulty in having men agree to employment in the forge shop.

Author's Closure

The authors wish to express their sincere appreciation for the instructive discussion which has been given this work.

In answer to Mr. Bender, it is true that much is unknown about the action of pulverized-coal burners. However, it has been fairly well established that turbulence does not appreciably affect the thickness of the gaseous film on the coal particle. In other words, high velocity does not scrub the products of combustion away so that raw-coal surface is made available for burning. Burning occurs not by vaporization but by diffusion through this tenacious film.

The only other value which turbulence can have is to distribute the air and coal uniformly throughout the furnace. This is the reason for the use of so-called turbulent burners in boiler furnaces. But designers do not depend too much upon this turbulence to get good distribution. They do not introduce all of the coal through a single, high-velocity, turbulent burner; instead they use a multiplicity of burners to distribute the coal and air across the furnace.

Mr. Grindle is an exponent of the total air burner for pulverized coal. Although the authors know of no steel-heating furnace where this is used, except on long high-temperature melting furnaces, it must be admitted that failures in its application to boiler furnaces and also theoretical considerations have discouraged experimentation with its use on small furnaces. The low velocities of flame propagation which obtain for lean mixtures seem to be against the introduction of all of the air mixed with the coal; some comparative data should be obtained on a small furnace using a total air burner and then the usual primary-secondary air burner.

Mr. Herrington's descriptions of actual installations are valuable in showing what can be done.

Mr. Pettibone's suggestion of the use of a simple compressed-air feeder is a valuable one. It will not work for all coals, but, for those with which it will work, it provides a very simple means of feeding coal. Just as in any powder-feeder, thorough drying is essential. However, with the sticking coals, referred to in the

paper, even completely dry coal will cause trouble. The action of such a feeder depends upon the uniform flow of coal toward the mixing chamber as a result of the slight pressure difference and gravitational force. The sticking coals are very apt to arch over the feeder, hang and drop, giving irregular feed. If some agitation device could be developed to mix completely and to keep mixed a small amount of air with the coal over the feeder, each particle would be surrounded by a tenuous film of air, and packing and sticking would not occur; the mixture would flow readily and uniformly.

As to feeding the coal-air mixture through $^1/_2$-in. pipe, the authors repeat that, for some coals, this is all right, but on others there will be plugging by the superfines. Brief tests made since presentation of the paper have shown that plugging is reduced when $^3/_4$-in. pipe is replaced by 1-in. pipe, thereby reducing the velocity. Apparently, plugging is caused by electrostatic charging of the superfines, some of which then are attracted strongly to the pipe wall; the lower velocity probably causes less charging, and hence less sticking to the wall. Also, the larger the pipe diameter, the less is the effect on resistance of a layer of coal adhering to the wall.

Preheating would of course effect savings and aid ignition. Past attempts to get high temperatures by means of alloy recuperators have been disappointing owing to short recuperator life. However, satisfactorily tight refractory recuperators have been developed. They are economically justifiable for large furnaces. For small ones they are less so. Nevertheless, as Mr. Pettibone implies, moderate preheating can be simply obtained by passing the secondary air through the slot stack. Time did not permit trying this on the forge furnace studied.

Unfortunately, there was not sufficient time to work on combustion controls of which Mr. Reaser speaks. Precise systems of combustion control have been developed for pulverized-coal-fired boiler furnaces, but automatic controls have been slow in coming to the forging field. Doubtless large savings and improved products will result from work along this line.

Mr. Rehfuss' remarks on the importance of low velocities verify the findings in the paper. However, this should not be taken to condemn controlled forced secondary air. The great virtue of natural draft here is that no amount of overzealous design can make a stack so powerful that it creates a blast sufficient to erode the refractories. Yet the blower does not necessarily have to blast the secondary air into the furnace either. There should be no mystery to the choice of fan and port areas so that velocities as low as desired can be obtained.

Although no question of the effect of different coals has been raised, some data which have been taken since that in the paper should be of interest. Table 4, herewith, lists the coals which have been tried, including those in Table 3 of the paper. Comparison was made on the basis of fuel required to maintain thermal equilibrium rather than on the basis of carbon loss, since these measurements were laborious and none too accurate.

The variation in coal requirement among the various coals was slight. Apparently, high specific surface of the low-volatile high-grindability coals offsets their slower ignition.

With all of the coals tested there was a gradual accumulation of ash dust on the hearth. Except when overheating, there was very little sticking of the ash. In actual practice, constant loading and unloading of the charge provides enough agitation so that all of this ash is carried off by the stack.

TABLE 4 PROXIMATE ANALYSES OF COALS TESTED[a]

	Elkhorn	Pocahontas No. 3	Pocahontas No. 4	Sewell	Moshannon	Island Creek
Moisture, per cent	0.2	0.1	0.2	0.2	0.3	0.2
Volatile matter, per cent	35.2	20.8	15.9	22.7	22.2	37.2
Fixed carbon, per cent	61.0	74.6	77.6	70.6	69.4	57.3
Ash, per cent	3.6	4.5	6.3	6.3	8.1	5.5
Sulphur, per cent	0.8	0.6	0.9	1.2	0.9	1.0
Heating value, Btu per lb	14850	14900	14880	14660	14260	14480
Ash-softening temperature,[b] F	2510	2300	2320	2430	2800	2850
Hardgrove grindability[c]	46–53	101–111	87–108	95	102–112	49–52
Percentage through 200 mesh	93	93	93	93	93	93
Specific surface,[d] sq cm per gm	2870	5280	5331	5695	5546	4200
Feed rate, lb per hr necessary to maintain temperature indicated, F	32	32[e]	33	28	28	32[e]
	2350	2350	2430	2310	2300	2350

[a] Basis—as fired.
[b] Average for mine as reported by Bureau of Mines.
[c] From "Grindability of Coals," Bulletin 3-241, The Babcock and Wilcox Co., New York, N. Y., 1938.
[d] Determined by Lea-Nurse air-permeability method.
[e] Estimated from tests in which equilibrium was not obtained.

The Corrosion of Stressed Alloy-Steel Bars by High-Temperature Steam

By H. L. SOLBERG,[1] A. A. POTTER,[2] G. A. HAWKINS,[3] AND J. T. AGNEW[4]

An investigation was undertaken at Purdue University of the corrosion by steam of various stressed samples of steels which are available for high-temperature service. Apparatus was constructed and techniques developed for measuring the amount of corrosion on stressed specimens. Data are presented in this paper, showing the effect of stress upon the corrosion of Carbon-Moly, 1.25 Cr-Moly, 2 Cr-Moly, 7 Cr-Moly, 9 Cr-Moly, and 18-8 stainless-steel samples. Within the range of the test conditions stress does not influence the penetration or corrosion due to high-temperature steam. No intergranular attack took place except for a small amount in the case of Carbon-Moly steel during a 2000-hr test. Chromium content increases the resistance of steels to corrosion by high-temperature steam.

An INVESTIGATION of the corrosion of unstressed low-carbon steel and various alloys by high-temperature steam has been carried on at Purdue University since 1934 (1, 2, 3).[5] In general, it may be stated that the resistance of alloy steels to high-temperature steam is greatly influenced by the amount of chromium present. Alloy steels containing 7 per cent or more of chromium showed good resistance to corrosion produced by steam at temperatures up to at least 1400 F. The 18-8 stainless steels were highly resistant to corrosion when subjected to steam up to 1400 F.

Since all of the tests had been conducted on unstressed specimens, it was decided to determine the effect of stress on the corrosion of bar samples of alloy steels by high-temperature steam, particularly with respect to intergranular attack. The results of this phase of the investigation are the subject of this report.

Test Apparatus

A schematic diagram of the apparatus is shown in Fig. 1. Steam from the laboratory header at 200 psi gage flowed through a solenoid-controlled emergency stop valve, a gas-fired steel-tube superheater, and an automatically controlled electric superheater to four vertical test sections operated in parallel. Steam from each section was condensed in a seamless copper-tube condenser and weighed. A starting condenser was provided to permit operation of the gas superheater without the flow of steam through the test sections. One thermocouple at the discharge end of the electric superheater controlled the power input to the electric superheater through a temperature regulator. A second thermocouple connected to an emergency regulator closed the steam valve and reduced the heater input in case of abnormal steam temperature.

The electric superheater was made of 95 linear feet of $^1/_2$-in. standard-iron-pipe size 18-8 stabilized stainless steel. The tube was wound into the form of a flat spiral and was embedded in a

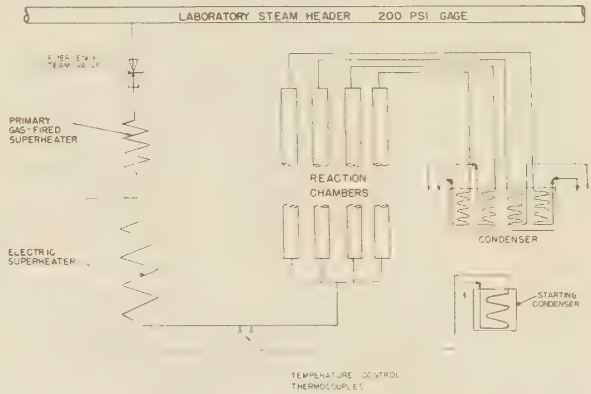

Fig. 1 Schematic Diagram of Apparatus

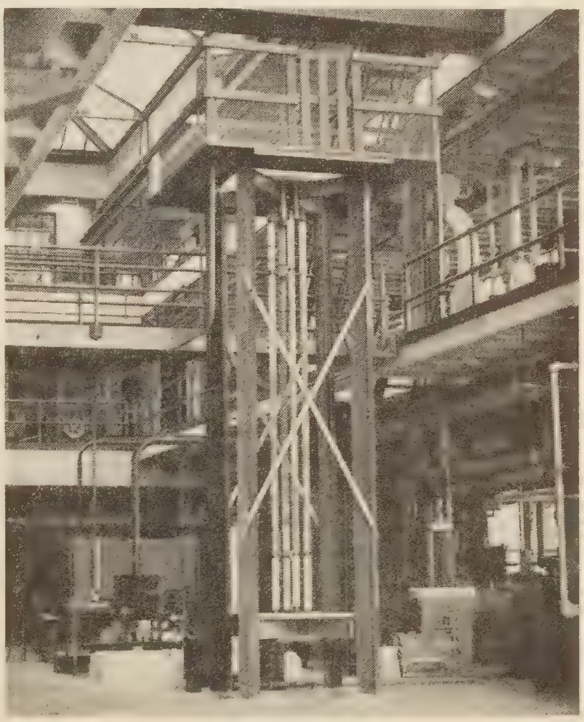

Fig. 2 View of Apparatus During Construction

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[5] Numbers in parentheses refer to the Bibliography at the end of the paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

thick layer of insulation. Stainless-steel lugs welded to the tube served as connections to the transformer.

Fig. 2 shows the four vertical test sections during construction. Each test section was made of 2-in. extra-heavy pipe 12 ft long. They were suspended from above in a framework consisting of four vertical I-beams with welded X-members at the top and near the bottom. Each test section was wound with an electric guard heater 2 ft long at each end and a main heater 8 ft long between the guard heaters. The power input to each heater was under separate manual control. The purpose of the heaters was to maintain a uniform temperature outside of the tube equal to the temperature of the steam flowing through the tube. Each test section was provided with a stuffing box at both the top and bottom, through which extension rods from the test specimens passed. The test specimens, occupying the middle 8 ft of each vertical test section, were suspended from the upper cross-member of the supporting frame and were loaded in tension from below the cross-member by means of calibrated steel springs.

Insulating key brick were laid up around the four test sections. The vertical insulated furnace was divided into four compartments, one for each test section, by brick laid in the form of a cross inside the cylindrical furnace shell. By means of this construction and different amounts of insulation on the piping which connected the electric superheater to the various test sections, it was possible to operate the four test sections simultaneously at different temperatures.

Outline of Investigation

Specimens of various diameters were machined from each of four selected steels of different analyses for each test. The specimens from each type of steel were coupled in series in the center of one of the four vertical test sections, suspended from above and loaded from below to a predetermined tensile load. They were then subjected to flowing steam at a predetermined temperature. Two series of tests were conducted. Test 1 was for 1030 hr, while test 2 was 2000 hr in length. Elongation, depth of corrosion, and tensile strength were determined and photomicrographic studies made.

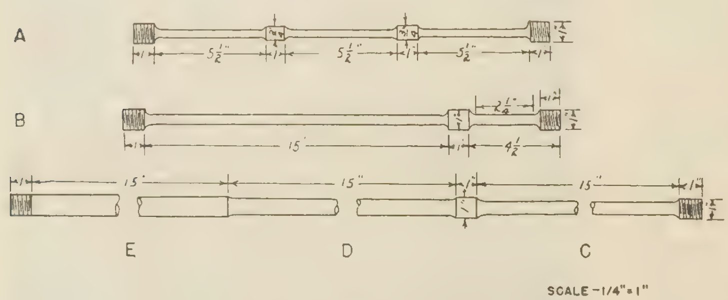


Fig. 3 Specimens for Test 1
(Duration of test, 1030 hr.)

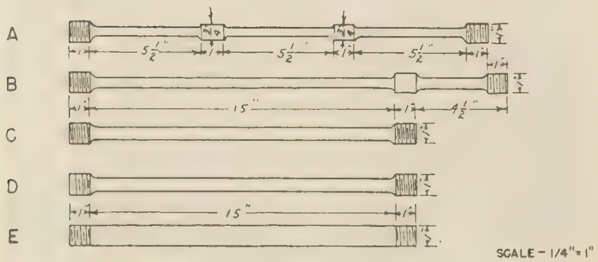


Fig. 4 Specimens for Test 2
(Duration of test, 2000 hr.)

Test Specimens

The specimens for test 1 were heat-treated and machined into sections as shown in Fig. 3. The Carbon-Moly, 1.25 Cr-Moly, and 2 Cr-Moly steels were annealed at 1550 F. The 7 Cr-Moly and 9 Cr-Moly steels were normalized at 1750 F and drawn at 1500 F (6 hr). The 18-8 steel was annealed at 1950 F and water-quenched. The short section of part *B* (Fig. 3) was used for determining the physical properties after the test had been completed. The specimens were connected together by means of threaded collars into which the ends of the sections were screwed. All specimens were coupled together in the order of progressively decreasing diameters. The largest section was placed in the upper part of the reaction chamber. All specimens were ground and lapped to final size. Due to the difficulty of annealing and machining the long section *C*, the design was altered in test 2 so that the specimen shapes for this test corresponded to those shown in Fig. 4.

The assembled bars were located in the center of the reaction tube by means of extension rods which were screwed into the couplings on each end of the assembled specimens. The extension rods passed through the packing glands at the top of the reaction chamber and were held in place by nuts. A thrust bearing placed on each extension bar separated the nut from the X-frame in order to eliminate the possibility of twisting the specimen. The lower extension rods passed through packed glands on the lower ends of the reaction chambers. A spring was then placed around each extension bar and held in place by a collar, thrust bearing, and nut.

The load on the assembled specimens was applied by tightening the nut on the lower extension bar, thus compressing the spring. By means of calibrated gage plates, the deflection of the calibrated springs could be accurately measured.

Method of Determining Extent of Corrosion

After the bar specimens had been ground and lapped, the diameters were determined by means of a micrometer. Slight variations in diameter along the length of the specimens were noted and a record made of the location of any irregularities. Gage marks were located on each specimen in order that the elongation during the test might be measured. A sample of each steel machined from the ends of the test bars was used for determining the specific weight of the material by means of a Jolly balance. After the test samples were taken from the reaction chamber, each bar was cut into three sections, each about 3 in. in length. The products of corrosion were removed from the test sections by an electrolytic stripping operation (2).

After stripping, the samples were accurately weighed and the dimensions determined. The original weight of the sample was calculated from the initial volume and specific weight. The difference in the initial and final weights of the sections were taken as the loss in weight of metal, from which the penetration was computed. This method of determining the loss in weight and depth of penetration from the initial volume, the elongation, and final weight of a specimen of known length is not as accurate as the method of weighing unstressed simple bar specimens before exposure to steam and after electrolytic stripping, as reported in earlier papers (1, 2, 3); consequently some scattering of results is to be expected.

TABLE 1 CHEMICAL ANALYSES OF STEELS INVESTIGATED

Steel	Test	Ladle analysis, per cent C	Mn	P	S	Si	Cr	Ni	Mo
Carbon-Moly	2	0.13	0.49	0.011	0.010	0.25	...	..	0.52
1.25 Cr-Moly	1	0.14	0.38	0.015	0.018	0.79	1.25	..	0.55
1.25 Cr-Moly	2	0.12	0.44	0.010	0.010	0.75	1.27	..	0.52
2 Cr-Moly	1	0.11	0.45	0.012	0.015	0.42	2.08	..	0.50
7 Cr-Moly	1 and 2	0.11	0.43	0.010	0.011	0.92	7.33	..	0.59
9 Cr-Moly	2	0.10	0.38	0.010	0.011	0.64	8.74	..	0.64
18-8	1[a]	0.07	1.14	0.079	0.081	0.36	18.81	9.04	..

[a] Bar analysis.

TABLE 2 PHYSICAL PROPERTIES BEFORE AND AFTER TESTING FOR STEELS USED IN TEST 1[a]

Steel	Remarks	Yield point, psi	Ultimate strength, psi	Elongation, per cent	Reduction in area, per cent	Brinell hardness
1.25 Cr-Moly	Before test	51700	77700	34.0	68.0	143
	After test	49310	75300	35.0	67.4	
2 Cr-Moly	Before test	42150	65100	38.0	75.7	121
	After test	38580	61670	35.0	75.5	
7 Cr-Moly	Before test	56810	81860	34.5	73.8	152
	After test	50440	83510	31.0	73.3	
18-8	Before test	32200	85130	64.8	70.5	208
	After test	68800	106970	43.0	60.6	

[a] Duration of test, 1030 hr.

TABLE 4 PHYSICAL PROPERTIES BEFORE AND AFTER TESTING FOR STEELS USED IN TEST 2[a]

Steel	Remarks	Yield point, psi	Ultimate strength, psi	Elongation, per cent	Reduction in area, per cent	Brinell hardness
Carbon-Moly	Before test	46580	62860	36.0	68.9	115
	After test	47050	62210	37.0	70.7	
1.25 Cr-Moly	Before test	57800	74570	35.5	72.2	143
	After test	47690	65630	34.5	69.1	
7 Cr-Moly	Before test	56810	81860	34.5	73.8	152
	After test	48360	82570	36.0	71.9	
9 Cr-Moly	Before test	45330	80640	34.0	73.9	152
	After test	50130	84910	30.0	72.4	

[a] Duration of test, 2000 hr.

The chemical analyses of the various steels tested are given in Table 1.

Test 1 of 1030 Hr Duration at Temperatures of 1130 to 1170 F

Specimens of 1.25 Cr-Moly, 2 Cr-Moly, 7 Cr-Moly, and 18-8 steels were placed in the reaction chamber and tested for 1030 hr. During this test three heater failures occurred, one packing gland blew out, and one power failure resulted which necessitated removal of the load from each series of specimens five times. In addition to these interruptions, the load was removed from the specimens at 229 hr because of the failure of specimen *A* of 1.25 Cr-Moly; at 235 hr because of the failure of specimen *A* of 2 Cr-Moly; and at 590 hr because of the failure of the 7 Cr-Moly sample *A*.

The physical properties before and after testing for the steels used in test 1 are presented in Table 2. The yield point and ultimate strength of the 18-8 stainless steel for the specimens, taken after the test, are considerably higher than for the specimens taken before the test. It is known that 18-8 steel is subject to strain hardening. In all probability strain hardening took place during the test, as a result of the several interruptions which caused the specimens to be cooled and heated and the stress relieved and reapplied. The variations for the other steels are not significant since considerable variation is to be expected because of the fact that only two tensile specimens were used to determine the physical properties before and after testing.

The corrosion results found for test 1 are shown in Table 3, together with the corresponding stress, creep values, and temperatures. Considering first the 1.25 Cr-Moly steel, it is apparent that the penetration does not increase as the stress increases. For the 2 Cr-Moly steel, the corrosion is higher for the higher stresses. In the case of 7 Cr-Moly steel, no relation exists between corrosion and stress. The corrosion product formed on the

TABLE 3 CORROSION RESULTS FOR TEST 1[a]

Steel	Specimens A: Stress, psi	Total average creep, per cent	Average temperature, F	Specimens B: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens C: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens D: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens E: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.
1.25 Cr-Moly	11300	Failure[b]	1132	6000	0.20	1147	0.00209	4200	0.16	1133	0.00306	3800	0.14	1131	0.00279	1700	0.00	1134	0.00362
2 Cr-Moly	9500	Failure[c]	1157	5400	3.22	1165	......	3600	0.29	1145	0.00249	2600	0.08	1143	0.00177	1300	0.00	1144	0.00174
7 Cr-Moly	7500	Failure[d]	1150	5200	2.62	1168	0.00103	3400	0.42	1157	0.00168	2500	0.15	1143	0.00156	1300	0.00	1143	0.00207
18-8	10500	0.01	1165	7800	0.00	1162	.....	5400	0.00	1158	.....	4900	0.00	1159	.....	2160	0.00	1163	.....

[a] Duration of test 1030 hr.
[b] Specimen removed after 229 hr due to fracture.
[c] Specimen removed after 235 hr due to fracture.
[d] Specimen removed after 590 hr due to fracture.

TABLE 5 CORROSION RESULTS FOR TEST 2[a]

Steel	Specimens A: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens B: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens C: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens D: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.	Specimens E: Stress, psi	Total average creep, per cent	Average temperature, F	Penetration, in.
Carbon-Moly	5000	0.71	1117	0.00451	3370	0.32	1122	0.00427	2340	0.13	1115	0.00376	1720	0.06	1110	0.00391	1000	0.00	1110	0.00376
1.25 Cr-Moly	5000	0.46	1167	0.00563	2710	0.14	1174	0.00719	1880	0.08	1142	0.00603	1380	0.05	1140	0.00509	810	0.00	1144	0.00307
7 Cr-Moly	6200	4.50	1167	0.00260	3360	0.25	1163	0.00233	2340	0.10	1153	0.00191	1720	0.04	1151	0.00261	1000	0.00	1150	0.00338
9 Cr-Moly	4030	1.02	1159	0.00194	2710	0.21	1156	0.00171	1880	0.06	1132	0.00197	1380	0.04	1140	0.00205	810	0.00	1161	0.00474

[a] Duration of test 2000 hr.

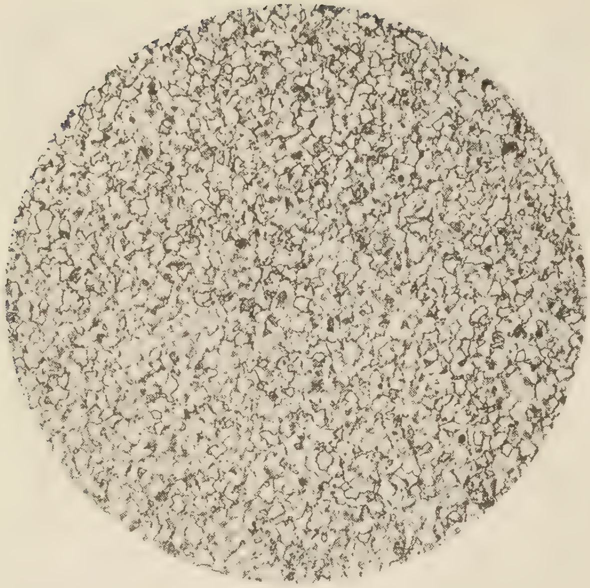
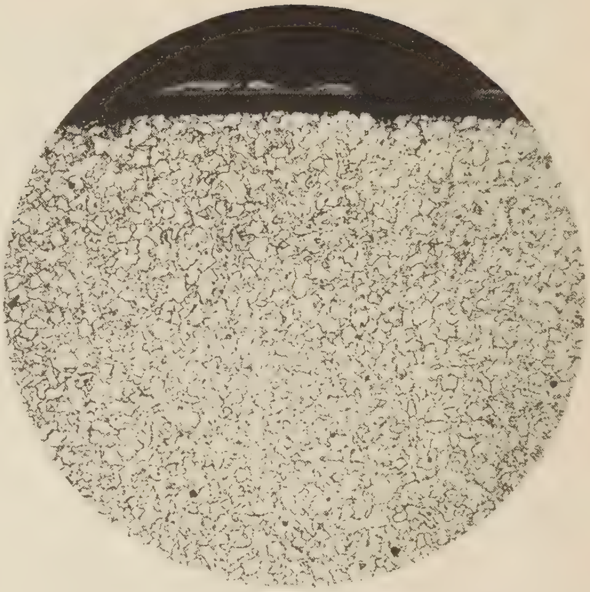


Fig. 5 Photomicrograph of Original Structure of 1.25 Cr-Moly: ×100



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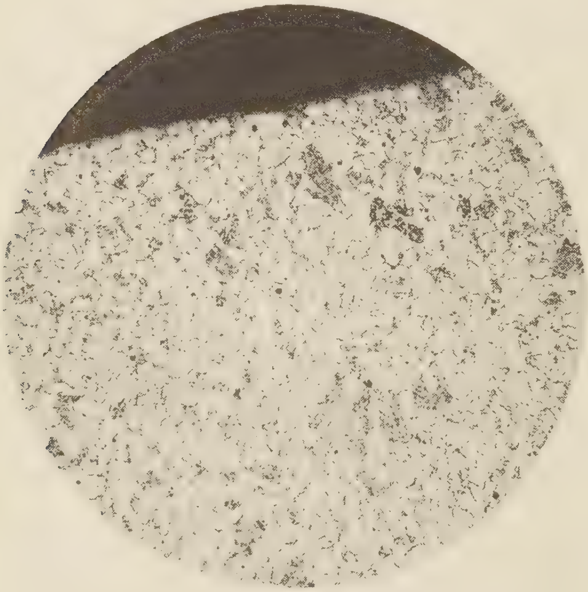
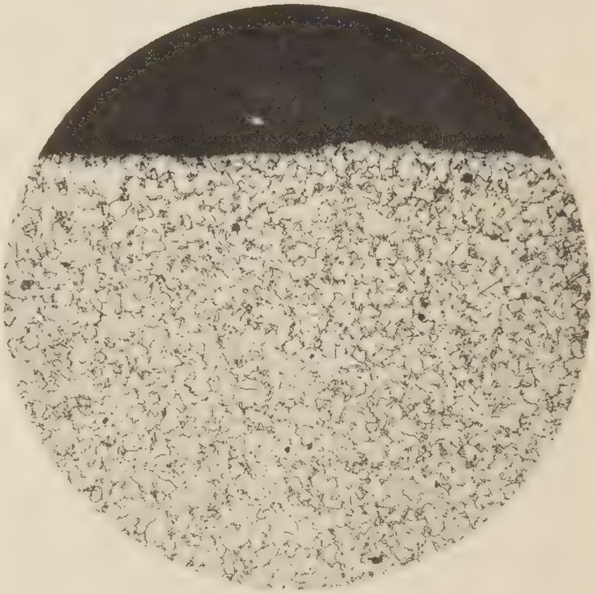


A



C

Fig. 6 Microstructures Showing Influence of Stress on Surface Attack of 1.25 Cr-Moly Steel After 1030 Hr; ×100
(Specimen *A*, stressed at 11,300 psi; specimen *C*, stressed at 4200 psi; specimen *E*, stressed at 1700 psi.)

18-8 stainless steel was too small to measure. From these results, there is no evidence to support the idea that stress increases the corrosion. No penetration values were cited for the three sections which failed because of the excessive amount of elongation which took place before failure.

The photomicrograph of the original structure for the 1.25 Cr-Moly steel used in test 1 is shown in Fig. 5. Photomicrographs of the structure after the test are shown in Fig. 6. Similar photomicrographs for the other steels were made. An examination of the photomicrographs reveals that no intergranular attack took place during the test. Some carbide precipitation occurred at the grain boundaries for the case of the 18-8 stainless steel. It may therefore be concluded that, for the time interval and temperature involved, stress did not affect the steam corrosion or the structure of the steels.

Test 2 of 2000 Hr Duration at 1110 to 1175 F

Specimens of Carbon-Moly, 1.25 Cr-Moly, 7 Cr-Moly, and 9 Cr-Moly steels were subjected to a steam atmosphere for 2000 hr.

The physical properties before and after testing for the steels used are shown in Table 4.

FIG. 7 PHOTOMICROGRAPH OF STRUCTURE OF 1.25 CR-MOLY AFTER 2000 HR; ×100
(Test 2: specimen *A;* 5000 psi stress; 1167 F.)

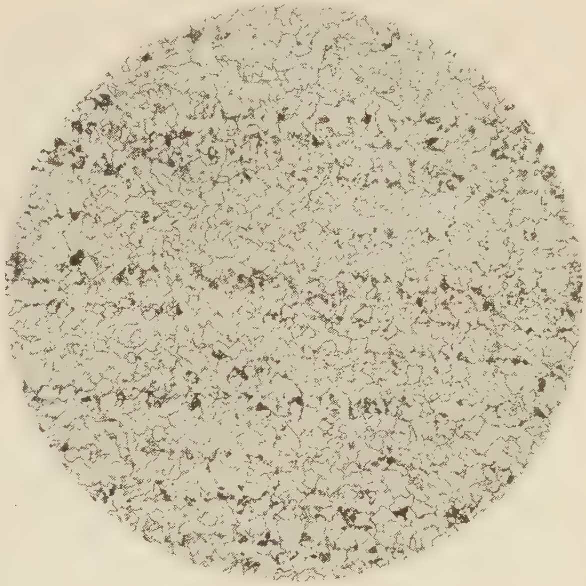


FIG. 8 PHOTOMICROGRAPH OF ORIGINAL STRUCTURE OF CARBON-MOLY; ×100

FIG. 9 PHOTOMICROGRAPH OF STRUCTURE OF CARBON-MOLY AFTER 2000 HR; ×100
(Test 2: specimen *A;* 5000 psi stress; 1117 F.)

The penetration results found for this test are shown in Table 5, together with the corresponding stresses, creep values, and temperatures. No relation exists between the corrosion penetration and the stress as shown from the results.

The photomicrograph for the original structure for the 1.25 Cr-Moly steel is shown in Fig. 5, and the structure after testing for the highest stressed sample is shown in Fig. 7. Similar photomicrographs were taken for the 7 Cr-Moly and 9 Cr-Moly steels; however, no structural change occurred. The original structure for the Carbon-Moly steel sample is shown in Fig. 8. The photomicrograph for the highest stressed sample of Carbon-Moly steel is presented in Fig. 9. This photomicrograph shows some of the remaining scale. It is very difficult to retain the scale layer during the preparation of the metallographic specimen. It will be noted that an attack has occurred in the first layer of grains for the Carbon-Moly steel. Other than the small attack shown for this steel, intergranular attack did not take place during the 2000-hr test. In the case of the 7 Cr-Moly steel, some surface roughening occurred.

Conclusions

For these tests the following conclusions may be reached:

1 The results of the 1030- and 2000-hr tests indicate that the stress, within the ranges of time, temperature, and stress used, does not influence the penetration or corrosion due to high-temperature steam for the steel tested.

2 No intergranular attack took place except for a small amount in the case of Carbon-Moly steel in the 2000-hr test.

3 Chromium content increases the resistance of steels to corrosion by high-temperature steam.

4 The 18-8 stainless steel is extremely resistant to steam corrosion.

Acknowledgments

This investigation was made possible through the assistance of The Engineering Foundation and various manufacturers of alloy-steel tubing. The authors appreciate the advice given by members of the A.S.M.E. Special Research Committee on Critical-Pressure Steam Boilers.

The authors are deeply indebted to Dr. C. L. Clark, research metallurgical engineer of The Timken Roller Bearing Company, Steel and Tube Division, for valuable assistance and constructive criticisms.

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Automatic Uniform Rolling-In of Small Tubes

By F. F. Fisher[1] and E. T. Cope,[2] Detroit, Mich.

While successful methods for rolling-in large tubes, such as boiler tubes, have been developed, the rolling-in of tubes in surface condensers, feedwater heaters, air coolers, air heaters, and oil coolers has been mentioned but never described in the literature of the art. This paper reviews some of the current methods of rolling-in small tubes in heat exchangers, pointing out the shortcomings of each. The authors have conducted extensive investigations on the procedures of rolling-in small tubes and have devised an entirely new technique which they describe. The method employs an electric-motor-driven parallel-rolling self-feeding expander, which is entered to its full length into the tube end; damage to tube end and tube sheet and excessive friction between tool and tube or tube sheet being prevented by appropriately shaped ball-bearing thrust collars. The degree of expansion is controlled by a current-limiting relay. The effect of the starting current, which is greater in value than the setting of the current-limiting relay and would cause it to chatter, is controlled by an auxiliary switch built into the mechanism of the expander-motor drive. No special training or skill is required in the application of this technique.

Introduction

The fastening of tubes into tube sheets, drums, and headers by expanding the ends or rolling-in, as this procedure is called among construction men, has been for many years the accepted practice in the installation of large tubes, such as in boilers and similar vessels and with the smaller tubes in feedwater heaters, air and oil coolers, and air heaters. On the other hand, it is only within the last fifteen years that the tubes in surface condensers have been rolled-in, and only within about five years that this method has been generally accepted. Previously, such joints were formed by means of various kinds of packing, each kind having certain advantages and disadvantages, but none as permanently satisfactory in operation or as low in cost as the rolled joint.

A close study of the technique for rolling-in boiler tubes became mandatory when operating pressures had increased to such values that the factor of safety of the joint was in some instances far below that of any other part of the boiler structure. Instances have occurred in which certain joints could not be made to withstand the Code test pressure during the initial inspection, not to speak of withstanding operating conditions. Two papers[3,4] by the authors of this paper, describing the results of their study of boiler-tube rolling, a short article[5] by Pollard, which was based on and confirmed the conclusions stated in the authors' paper,[3] and a McGraw prize paper[6] by one of the authors of this paper are, as far as is known, the only published articles in English dealing with the art and science of tube rolling. No other detailed information is available pertaining to the rolling-in of small tubes.

It is true that the rolling-in of condenser tubes was mentioned frequently in reports of the Prime Movers Committee of the late National Electric Light Association and of a similar committee of the Edison Electric Institute, but nowhere will one find a description of the procedure by which satisfactory joints can be assured. In this paper the authors will discuss some customary practices and will tell of the development of a new technique by the use of which the small tubes installed in heat exchangers can be automatically rolled-in to produce joints of uniform strength and stability. This procedure eliminates all the guesswork inherent in the rolling of joints by customary methods. Once the best joint conditions have been determined experimentally (using a given tube and tube-sheet combination), these conditions can be duplicated, so joints can be produced by anyone at a rate equal to or in excess of that attainable by the use of the less accurate method now employed. The technique of applying this new method also compensates for the elongation incident to all tube rolling, leaving the newly rolled tube in a condition in which there is little if any residual axial stress.

General Discussion

A consideration of the rolling-in of large and of small tubes will make it apparent that the sequence of events that occurs in the rolling-in of small tubes is identical with that occurring with large tubes, as described by the authors.[3,4] A brief résumé of this description will not be out of place here. In order to allow for easy insertion of the tube into the tube hole, the tube hole is made slightly larger in diameter than the outside of the tube. Thus, during the expanding of the tube, the joint does not begin to exist until the tube has been stretched into contact with the hole. From this point on, expansion of the tube radially is restrained by the tube-sheet metal, the tube-wall metal being forced to flow axially. The amount of this axial movement, commonly referred to as "elongation," bears a fairly definite relation to the strength and stability of the joint and is therefore used as a measure of the degree of expansion imparted to the joint. The metals of the tube wall and of the tube sheet immediately adjacent to the tube hole are cold-worked by this action and an increase in hardness occurs with a corresponding decrease in ductility and shock resistance. Because of this, the cold-working of these metals should be held to as small a value as is necessary to produce joints of the required strength and stability and preferably should be uniform throughout the length of the joint.

The elongation subjects the tube to an axial compressive loading that is transferred to the tube sheets as a force tending to bend or separate them. In any bundle of tubes, the forces resulting from rolling-in the tubes may produce a very complex condition of loading of the tube sheet, unless a carefully worked out sequence of the rolling operation is followed, and the tubes

[1] General Foreman of Mechanical Erection, Construction Bureau, The Detroit Edison Company.

[2] Research Department, The Detroit Edison Company. Mem. A.S.M.E.

[3] "Rolling-in of Boiler Tubes," by F. F. Fisher and E. T. Cope, Trans. A.S.M.E., vol. 57, 1935, p. 145.

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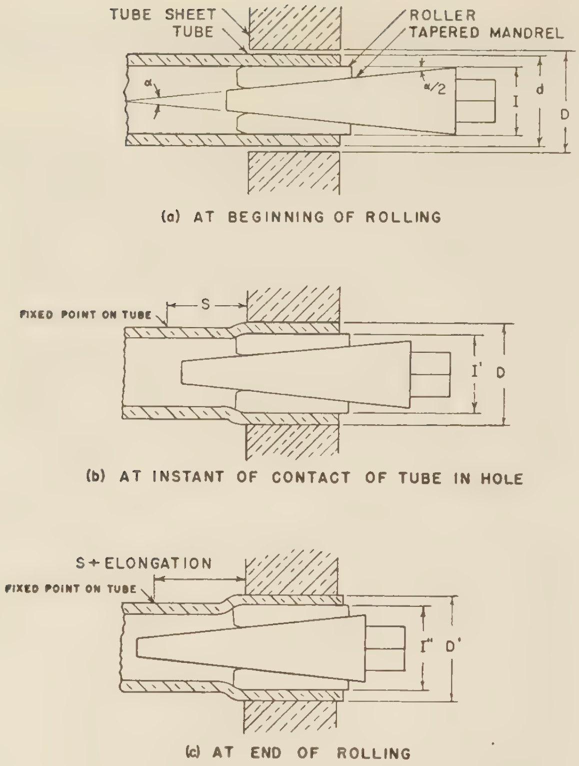

Fig. 1 Non-Self-Feeding Expander

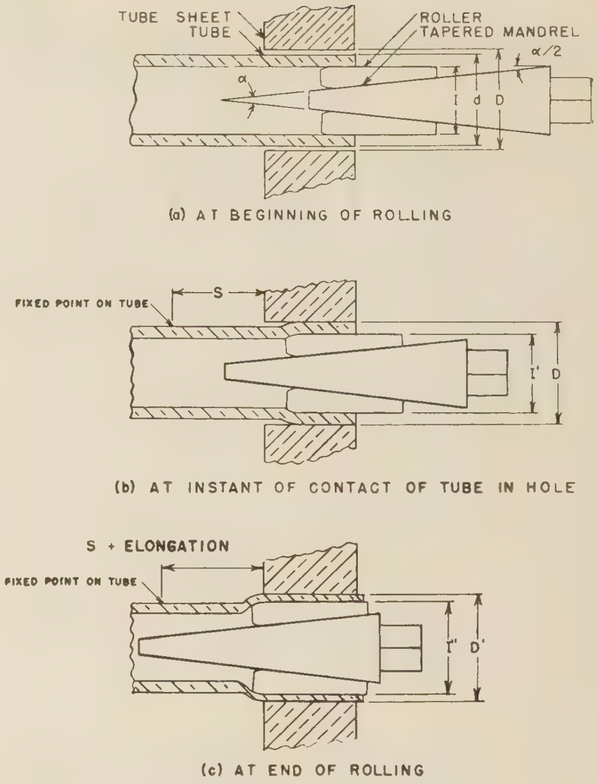

Fig. 2 Self-Feeding Expander

are rolled to uniform elongation. The occurrence of ruptured tubes that have failed because of excessive axial loading probably attended by vibration is not uncommon, especially in feedwater heaters where thick-wall tubes designed to operate at relatively high temperatures and pressures are used.

The authors will show that the amount of the axial stress in small rolled tubes can be eliminated completely or held to low values by the use of the new technique, which is the subject of the paper. Such control will also reduce and largely correct the complexity of the loading imposed by the tubes on the tube sheets. In order to gain a clearer understanding of the basis for this claim, brief descriptions will be given of the operation and use of the tube expanders usually employed.

The Expanding Tool

The expanding tools used on small tubes, shown schematically in Figs. 1 and 2, usually consist of three rollers supported by a cylindrical basket or cage forming the body of the tool. The rollers are placed at intervals of 120 deg around the axis of the body. A tapered mandrel extends axially through the tool body and contacts the rollers. The larger end of the mandrel is fitted to a wrench or to an air or electric motor. By these means the mandrel is revolved about its axis. This action revolves the individual rollers in a direction opposite to that of the mandrel, carrying the cage in the same direction as the mandrel but at a reduced rate. Such tools are of two types, i.e., the older non-self-feeding type, having the axis of any roller in a plane passing through the axis of the mandrel, and the self-feeding type, having the axis of each roller set at an angle (feed angle)[7] of from $1^1/_4$ to 2 deg to a plane passing through the axis of the mandrel. The taper of the mandrel varies from $^1/_4$ to $^1/_2$ in. per ft.

[7] Fig. 6 shows this construction.

The rollers of either type of tool may be cylindrical or tapered. If cylindrical, the diameter of the joint will be largest at the outside of the tube sheet. If tapered, the taper of each roller may be one half that of the mandrel, a condition frequently encountered. The finished joint is practically cylindrical. This second type, the so-called "parallel expander," is the one the use of which is strongly advocated by the authors and has been used in all tests reported herein. The reason for favoring this tool is that the use of it results in uniformly cold-worked tube walls and tube sheets and produces joints of approximately equal strength in either direction axially.

Rolling-In a Joint by Use of Non-Self-Feeding Expander

The non-self-feeding expander is essentially a hand-operated tool and is not suitable for production work. When in use, the tool is entered to its full length into the tube, and the mandrel is forced forward by being driven with a mallet. The tool is revolved until the tube has been enlarged enough to offer little resistance to further rolling and then the mandrel is again driven forward. Again the tool is revolved as before and the operation is repeated until the operator considers the joint to be complete.

The action will be understood better by the following description and by reference to Fig. 1. In this illustration, the clearances, changes in dimension, and the angles of the tapered mandrel and rollers are all greatly exaggerated, in order to bring out the details of the action of the tool. The two rollers shown are normally 120 deg apart, but in Fig. 1 one roller is oriented into the plane of the sketches to lend clarity.

Fig. 1(a) shows the position of the non-self-feeding tool at the beginning of the operation of rolling-in a tube. The nominal clearance $D - d$ depends upon the particular choice of tube and tube hole. The initial tube-wall thickness, $(d - I)/2$, changes to

$(D - I')/2$ at the instant the tube contacts the tube hole, as in Fig. 1(*b*), and to $(D' - I'')/2$ when the rolling-in is completed, as shown in Fig. 1(*c*). The angle of the tapered mandrel is α and that of the roller $\alpha/2$, so that the outside surfaces of the rollers during the operation of the tool generate a cylinder.

Rolling-In a Joint by Use of Self-Feeding Expander

The three important positions of the self-feeding expander during the operation of expanding a tube are shown in Fig. 2. In this illustration, as in Fig. 1, dimensions are exaggerated, and the symbols and letters have the same significance as in the former figure. In Fig. 2(*a*), the expander has been introduced into the tube by an amount found by trial to be about correct which, in the rolling techniques usually employed, is determined largely according to the judgment of the workman. This judgment is very largely a matter of guesswork with a basis of experience. In Fig. 2(*b*), the tool has advanced into the tube by virtue of its self-feeding characteristic and a portion of the tube has been expanded into contact with the hole. The joint has now just started to come into existence. Further operation of the expander results in the condition shown in Fig. 2(*c*). Here, the expander tool is "clear home" and has produced a joint of the desired length but of uncertain quality. Upon the skill of the operator depends the complete and correct expansion of the joint, with contact throughout the desired length of tube. The self-feeding expander, in spite of the necessity for a highly skilled operator, is the most satisfactory tool available. This type of tool is referred to whenever an expander is mentioned throughout the remainder of this paper.

Methods of Control

Uniform-Entrance Method. To eliminate the uncertainty inherent in the use of the self-feeding expander, several methods of control have been devised. The uniform-entrance method is one of these. In this, the rolling operation is controlled by the distance that the mandrel enters the body of the tool. Although this method produces tube ends of uniform internal diameter, it is obvious that uniformly expanded joints can be produced by this method only if the tube holes are of identical diameter and the tubes have the same inside diameter and thickness of wall. Such uniformity is seldom found in actual tubes and tube holes; it is customary in rolling-in tubes in an actual tube sheet, first, to measure a number of tube holes and tube ends taken at random and then to set the mandrel stop to allow a definite mandrel travel to suit the average conditions found. The amount of mandrel travel is determined by trial and experience and, because of variations in mandrel size, is peculiar to each tool used.

It is apparent that the chance combination of (*a*), the largest tube-sheet hole and the tube having the thinnest wall, or (*b*), the smallest tube-sheet hole and the tube having the thickest wall, would result in a great variation in the condition of the joints rolled by this method. This will be clear when it is considered that, in condenser practice especially, the difference between a joint that is properly rolled and one that is underrolled or overrolled is a matter of a few thousandths of an inch.

Elongation Method. This method, as applied to larger tubes, has been fully described in the authors' papers[3,4] on boiler-tube rolling previously mentioned. It is believed to be the most scientific method yet proposed. In this technique, the tubes are rolled to a predetermined elongation measured individually as the work proceeds and, for this reason this technique is not readily applicable to the rolling-in of large numbers of small tubes because of the labor involved.

Current-Input Method. An improvement in the technique for rolling-in small tubes was described in the McGraw prize paper[6] previously mentioned. In this the completion of the rolling-in

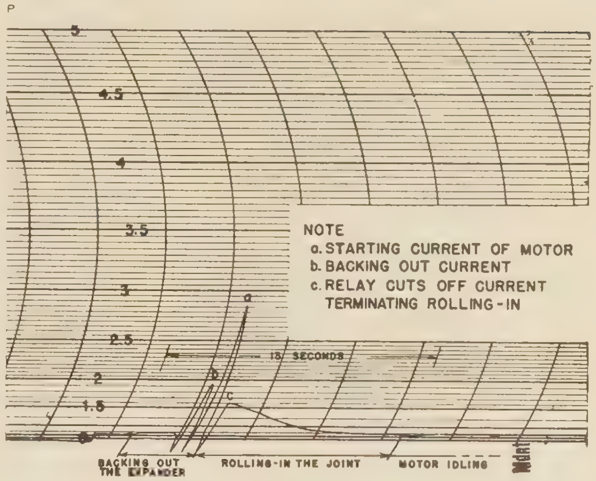

Fig. 3 Recording-Ammeter Record Taken During One Rolling-In Cycle

of a joint was reported to have been indicated by the rate of current input to the expander motor. An automatic current-limiting relay connected into the motor circuit controlled the rate of input, cutting off the supply when a predetermined value of current had been reached. This value had been determined by tests on sample joints and was such that uniformly and properly rolled joints were produced between tube holes of usual commercial variations in diameter and tubes of commercial variations in diameter, hardness, and tube-wall thickness. The expanding technique used with the current-input method of control was identical with that described under "Rolling-In a Joint by the Use of the Self-Feeding Expander."

The fact that the use of this control method resulted in the production of joints that were more uniformly rolled than had been possible by older methods is shown by laboratory tests and by the results obtained in the case of two condensers recently installed. In these two condensers only 27 joints in a total of 44,000 showed signs of leakage and had to be rerolled.

The main objection to this technique as reported was found to be that the output of rolled joints was less per man-hour than that obtained by the use of the less accurate methods employed at that time. The reason for this was that the line voltage of 116 v had to be reduced by one half in order that the relay employed would operate in a satisfactory manner. Fig. 3 is a recording-ammeter record of one rolling cycle, showing that the current required to start the motor and to back out the tool was greatly in excess of the relay setting, causing the relay to chatter. This interfered with rapid production of joints. A second shortcoming of this technique was that the relay might operate, giving the indication that the joint had been finished when in reality none or an insufficiently rolled one had been produced. This would occur when the tool body came into contact with the tube end or tube sheet, and a part of the input to the motor was dissipated in friction instead of in expanding the tube. In order to avoid this condition, considerable skill was required in handling the tool and, in many instances, it was necessary to reintroduce the expander two or three times into the tube end before the rolling-in was completed in a satisfactory manner.

Improved Rolling Technique

A comparison of the different rolling techniques and the methods of control showed that the current-input method possessed the greatest number of advantages with the fewest shortcomings.

It would be of great advantage if a technique could be de-

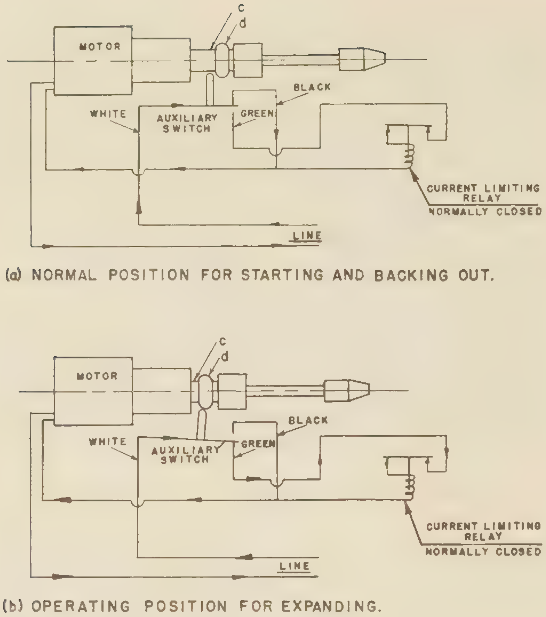

Fig. 4 Electric Circuit for Expanding Tool

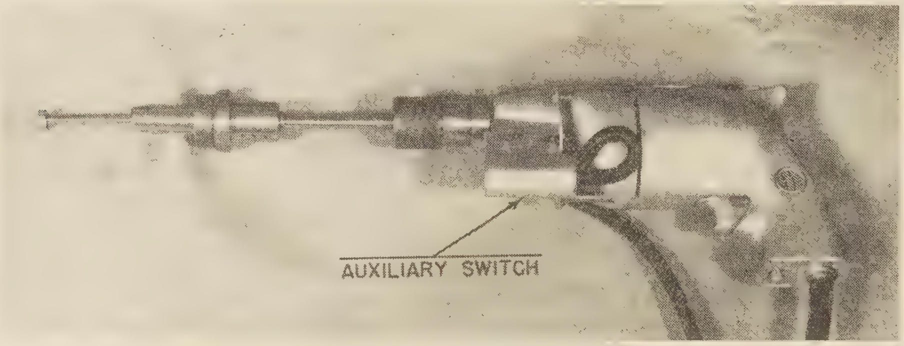

Fig. 5 Expanding Tool With Auxiliary Switch

veloped, by the use of which the output of rolled joints per man-hour would equal or exceed that possible by the use of other methods without sacrificing the uniform quality of rolling. Also, this procedure should be one in which a man of average ability and without previous training would simply insert the expander a predetermined distance into the tube and operate it until the control device indicated that the joint was fully expanded. This would call for an expanding technique similar to that employed with the non-self-feeding type of expander as shown in Fig. 1, but employing the self-feeding properties of the expander shown in Fig. 2. The degree of expansion would be controlled automatically, preferably by shutting off the power supply to the machine at a predetermined value of current. This is most accurately and conveniently done in an electric-motor-driven expander by the use of a current-limiting relay.

It is true that many of the electric-motor-driven mechanisms used for operating tube expanders are equipped with adjustable friction clutches so that they release on overload. Experience has shown that these devices are not as flexible or dependable as the current-limiting relay. This is true because the friction clutches are subject to wear and the adjustment of the control is characteristic of each individual motor drive and therefore cannot be specified; that applying to one tool does not necessarily apply to another. On the other hand, once the power input, over that of the unloaded-expander drive, that is necessary to produce joints of certain characteristics has been determined for a given set of conditions, similar joints can be produced at will anywhere by the use of a current-limiting relay similarly set, if the same conditions obtain.

In order to utilize the advantages of the current-input method as a foundation on which to develop the ideal technique, efforts were made to correct the shortcomings of the former. The first of these shortcomings, the low rate of output, resulting from the use of the particular current-limiting relay available, was corrected by the introduction of an auxiliary switch.

Auxiliary Switch. The auxiliary switch was mounted on the tool in such a manner that it was actuated by and interlocked with the operation of the expander tool so that it automatically disconnected the control relay during the starting of the motor and during the backing out of the expander, connecting it only during the time that the tool was expanding a tube. It is to be noted that the driving motor runs in one direction only and the change in direction of rotation of the tool is accomplished by means of gears and clutches built into the motor. With such an auxiliary switch mounted on the driving mechanism, the sequence of the rolling operating is as follows.

The auxiliary switch normally connects the motor circuit to the line and at the same time disconnects the relay. This is shown diagrammatically in Fig. 4(*a*), the circuit being completed from the "line" to "white," through "auxiliary switch" to "black," and thence to the motor. The spindle *c*, Fig. 4(*a*), is normally held in this position by a spring. The direction of rotation is that which normally backs the tool out of the tube. The expander is introduced to the desired depth into the tube end, and a slight forward pressure is exerted by the operator on the tool; this pressure shifts the position of the spindle *c* so that the collar *d* actuates the auxiliary switch, establishing the connections shown in Fig. 4(*b*). This establishes the motor circuit through the relay so that the current flows from the "line" to "white," through the "auxiliary switch" to "green," through the relay to the motor. The axial movement of the spindle also reverses its direction of rotation by shifting the gear connections to the direction that feeds the tool into the tube, thereby expanding it. This action continues until the current reaches such a value that the relay operates to cut off the current supply to the motor, thus terminating the rolling-in of the tube.

A slight outward pull on the tool simultaneously re-establishes the electric circuit to the motor, as shown in Fig. 4(*a*), starting the motor, and reverses the direction of rotation of the spindle *c*, thereby backing out the tool. If the highest rate of rolling-in of

joints is to be attained, the relay should be quick-acting and the time required for the automatic resetting of the relay should not be greater than that necessary for backing the tool out of the tube. This arrangement, employing the auxiliary switch, has made possible the operation of the expander on normal line voltage, removing completely the need for using half voltage, and has reduced the time necessary for rolling-in a joint to about one half of that required when the lower voltage was employed, making the production of joints at least as rapid as when the customary method was used.

A motor drive, equipped with auxiliary switch, is shown in Fig. 5.

Thrust Collars. In order to avoid the second fault, that of possible contact between the tool and the tube end or tube sheet, which causes an increase in load on the motor drive because of friction between the body of the tool and the tube or tube sheet, and at the same time to avoid the necessity for repositioning the expander several times to a more advantageous starting point in the tube, the expander was entered to its full depth into the tube before starting the rolling-in operation, as specified under "The Improved Rolling Technique." This position of the self-feeding expander prevents the normal feeding of the tool into the tube but the self-feeding tendency is still present. This tendency, if not circumvented, would bring the tool body into forcible contact with the tube end or the face of the tube sheet, resulting in damage to one or the other unless means were provided to protect these parts. These protective means were provided in the form of ball-bearing thrust collars having an inside diameter sufficient to clear the rollers at their greatest spread. These are shown in Fig. 6. The ball thrust bearing was chosen because of its constant low-friction characteristic which makes it peculiarly suitable for use in connection with control by means of the current-limiting relay.

A study of the conditions found at the two ends of the tube led to the conclusion that the best result could be obtained if a different type of collar were used on each end. It is considered good practice to roll-in the inlet end of all tubes first so that the entrance bells or flares may be formed uniformly. In the condenser-tube-rolling standards adhered to in the company with which the authors are associated, this end of the tube is made flush with the face of the tube sheet. This is shown in Fig. 6(*a*). The bell is formed with a belling tool after the tube has been rolled-in. If, at the beginning of the rolling-in operation, the tube end is below the surface of the tube sheet, the action of the tool will automatically draw the tube into the desired position. If conditions make it necessary that the position of the tube end should be different from that shown in Fig. 6(*a*), the necessary change in the thrust collar would accommodate this position whether it were below the face of the tube sheet or beyond it.

The discharge end of the tube presents a set of conditions that are entirely different from those found at the inlet end because of the fact that variation in the length of the tube or variation in the distance between the outside faces of the tube sheet may result in different lengths of overhang. This at most amounts to a small fraction of an inch. If trimming of the tubes is stipulated, it is not done until after the rolling-in has been completed in order to avoid the inclusion of cuttings in the rolled joint.

In order to provide for the condition of overhanging tube ends in rolling-in the outlet ends of the tubes, an expander having a thrust collar with skirt is used, as shown in Fig. 6(*b*). The internal dimensions of this skirt are such that the tube is cleared radially and axially. The same expander may be used in rolling-in the outlet end as was used on the inlet end.

When the expander is inserted into a tube until the skirt touches the tube sheet, its operation will produce a joint of uniform predetermined length regardless of the length of the tube extending beyond the tube sheet, if that length is not greater than the internal depth of the skirt. The length of the overhanging end of the tube has been found to have an insignificant effect on the power requirement of the tool, and so does not affect the uniformity of rolling when this is done by the automatically controlled current-input method.

Practical Application of the Improved Technique

The use of the thrust collar, in connection with the introduction of the tool to its full depth at the start of the rolling-in operation, corrected the faults mentioned and dispensed with the need for skilled operators. This arrangement plus the use of the expander tool in combination with the auxiliary switch, already described, and having a current-limiting relay in the motor circuit, makes it possible to meet the requirements of the ideal tube rolling-in technique, as stated under "The Improved Rolling Technique," viz. " . . . a technique . . . by the use of which the output of rolled joints per man-hour would equal or exceed that possible by the use of current methods without sacrificing the uniform quality of rolling. This procedure would be one in which a man of average ability and without previous training would simply insert the expander a predetermined distance into the tube end and allow the tool to operate until the control device indicated that the joint was fully expanded."

Travel Limit. It has been found expedient to make it impossible for the mandrel to enter the tool body so far that the rollers come in contact with the inside of the thrust collar. It is of course evident that the inside diameter of the thrust collar should be large enough to permit the rolling of tubes in holes of proper size yet not large enough to admit the end of the tube. If contact between rollers and thrust collar should occur such contact would impose a load on the motor sufficient to operate the relay even though the joint would not be formed properly. Such a condition would occur if the tube hole were seriously oversize and it would appear by the action of the tool that a satisfactory joint had been made when in reality none had been produced. In order to avoid such an occurrence and to provide external evidence to the operator that unusual conditions exist, a travel limit, shown in Fig. 6, is fitted to the mandrel. This is a sleeve so placed on the mandrel that the sleeve comes in contact with the body of the

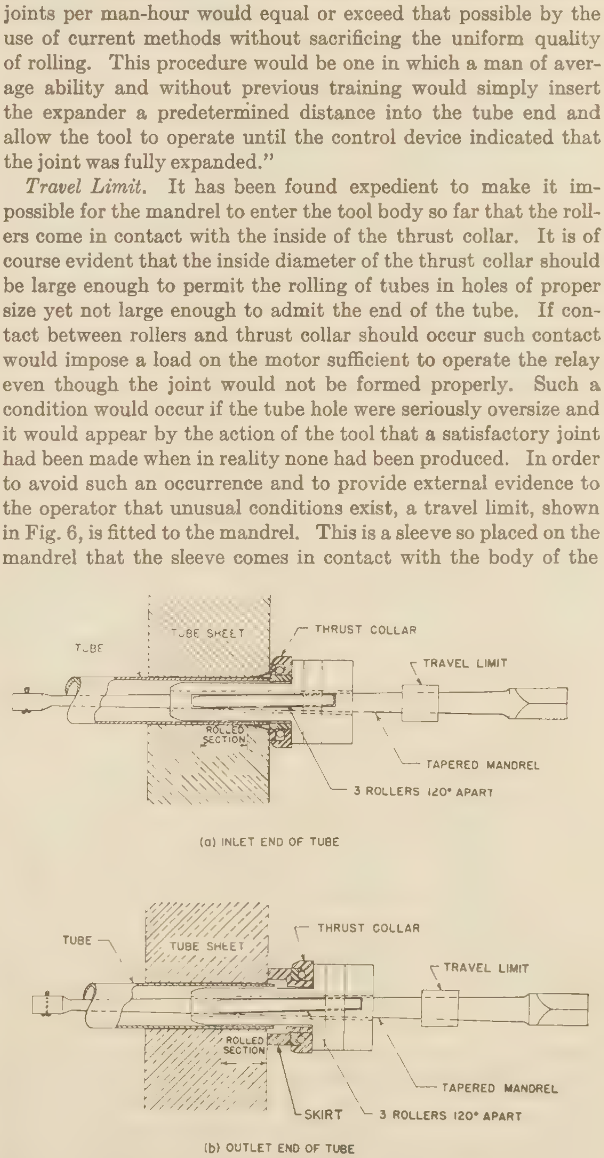

Fig. 6 Assemblies of Expanding Tool for Expansion of Inlet End and Outlet End of Tube

tool before the rollers come in contact with the thrust collar.

Compensating for Elongation. The use of the thrust collar with a skirt for the rolling-in of the outlet end of the tube makes possible an innovation in tube rolling, namely, compensation for the elongation incident to all tube rolling. It has long been known that, when both ends of a tube are rolled into fixed tube sheets, the elongation of the second end rolled subjects the tube to axial compression. The compression depends on the amount of elongation and sets up an axial thrust which often results in bowing of the tube, thus displacing it and impeding the flow of steam or air through the passes. Instances have come to notice in which the failure of small tubes could be accounted for only on the assumption that the tubes were under severe axial stress resulting from overrolling or nonuniform rolling. The axial thrust caused by the elongation in the tube often results in marked deformation or displacement of the tube sheet. This can be corrected only if a carefully planned sequence of rolling the tubes in a given tube bundle is observed. It is obviously very desirable that the tube be in an unstressed condition at the completion of the rolling-in process.

In order to achieve this condition, it is necessary that the elongation of the tube resulting from the rolling-in be offset in some manner. Elongation occurs in all tube rolling and cannot be prevented, but there seems to be no reason why it cannot be compensated for. The authors, in a previous paper,[4] proposed one solution to this problem. They suggested that, by the application of heat, the tube could be lengthened by an amount equal to the specified elongation, then rolled to that elongation. When cooled, the rolled-in tube would not be under axial stress. The same result may be achieved on small tubes during the rolling-in operation by use of the self-feeding expander equipped with a thrust collar having a skirt as described. This desirable condition occurs automatically as will be shown.

Let it be assumed that a tube has been rolled-in at the inlet end and that the outlet end is ready for rolling. The expander, equipped as shown in Fig. 6(*b*), is entered into the unrolled end until the skirt is in contact with the tube sheet, and rolling begins. The skirt, being in contact with the tube sheet, prevents the expander from feeding into the tube as would normally happen if no stop were present. Instead of feeding into the tube, the expander tends to draw the tube outward. But the tube is held rigidly at its opposite end, and the action places the tube under tension by a force equal to the axial pull of the tool. This stretches the tube measurably. The amount of this stretching appears to depend upon the feed angle of the expander, the number of rollers, and the length of the rollers in contact with the tube. The ideal combination of these factors would be attained if the stretching of the tube were exactly equal to the desired elongation, and completion of the rolling-in of the joint with its incident elongation would leave the tube in an unstressed condition axially. That this has been virtually accomplished will appear from the results of the following tests.

TABLE 1 RESULTS OBTAINED IN ROLLING-IN FOUR CONDENSER-TUBE JOINTS[a]

(Tubes were $^3/_4$ in., 18 gage, 21 ft long)

Joint no.	Stretching of tube, as shown by movement at end being rolled-in, in.	Elongation, in.	Net change in length of tube,[c] in.	Value of current at end of rolling-in, amp	Push-out strength, lb
1	0.0070	0.0098	+0.0028	1.7	2250
2	0.0111	0.0087	−0.0024	1.7	2250
3	0.0072	0.0100	+0.0018	1.7	2450
4	0.0133	0.0103	−0.0030	1.4[b]	2050

[a] Length of joint $^{15}/_{16}$ in; feed angle of tool 1° 30′.
[b] The correctness of this reading is questioned.
[c] A plus (+) net change in length of tube represents a compressive final loading, and a minus (—) net change in length represents a tension final loading of a tube.

Test for Validity of Compensation for Elongation

In order to prove the correctness of the theory, that compensation for elongation would leave the tube in an unstressed condition axially, a series of tests was conducted on a $^3/_4$-in. 18-gage condenser tube held between tube sheets 21 ft apart. The necessary readings were made to check the stretching of the tube during the rolling-in up to the instant of contact between the tube and tube hole and the effect of elongation in compensating for this stretching. The results of these tests are shown in Table 1.

The results shown in Table 1 were obtained by an operator who had never rolled-in a tube in an actual condenser and had none of the training or skill usually found necessary with the other methods of rolling. There was no need of any skill. All that was required was to make sure that the tool was properly inserted into the tube before the operation was begun and to reverse the expander when the relay operated. Point *c* in Fig. 3 shows the value of current at which rolling-in is terminated.

An examination of the results shown in Table 1 reveals that in two of the four cases the stretching of the tube during the rolling-in more than compensated for the elongation. In all cases there was little residual stretching or axial compressing of the tube. If the four joints tested had been successive or contiguous in the tube sheets of a heat exchanger, the net load on the section of tube sheets to which they were attached would have been small indeed, and the ideal condition of unstressed tubes and tube sheets would have been approached.

Rough quantitative checks obtained recently in rolling-in a portion of the tubes in a large condenser confirm the results of these tests.

The study of this method of tube rolling has not been extended by the authors to include other values of clearance between tube and hole, the use of tools having different feed angles, or joints of different length. Time did not permit such an extended study. It is hoped that other investigators may undertake to study the effect of variations in these items.

Summary

This paper reviews some of the current methods of rolling-in small tubes such as are used in surface condensers, feedwater heaters, air and oil coolers, and air heaters, pointing out the shortcomings of each method. It emphasizes the fact that an operator who undertakes to roll-in joints by any of these methods must be well trained and must possess considerable skill in using the expander tool. Yet, even in the hands of the most expert operator, uniform joints of optimum rolling cannot be assured. The elongation incident to rolling is always present, as is also the resulting condition of axial loading of the tube which produces complex loading and deformation of the tube sheets. A carefully planned sequence of rolling must be applied in order to distribute this loading in such a manner that tube failures may be avoided.

An entirely new technique of rolling-in tubes is described. This employs the familiar tools with the addition of appropriate thrust collars. The tool is entered to its full depth into the tube at the beginning of the operation and the current-limiting relay automatically stops the rolling by shutting off the current to the motor when the joint has been expanded by the correct amount as determined previously by laboratory tests on sample joints. This relay is not in the circuit during the starting of the motor or during the backing out of the tool, a very important new development. In this technique the elongation is compensated for in whole or in part by the stretching of the tube during the first stage of the operation, that in which the tube end is enlarged to fit the tube hole.

Results obtained from tests on rolled joints involving a $^3/_4$-in. 18-gage condenser tube 21 ft long are given. In these, the net final uncompensated axial stretching of the tube in the joints studied was about one fourth of the normal elongation. The resulting force imparted to the tube sheets would be very small as compared with that normally imposed by the other methods of tube rolling.

This new technique is independent of the skill and training of the operator so that any normal person with no previous experience in tube rolling can apply it to produce joints that are rolled uniformly to the desired degree at a rate equal to or greater than that attainable by the use of the other methods.

While it is true that this technique has been applied only to the rolling-in of condenser tubes, there appears to be no valid reason why it cannot be applied successfully to the rolling-in of the small tubes in all types of heat exchangers.

Discussion

J. F. Grace.[8] In the rolling operation of Worthington Pump and Machinery Corporation, the tool head is inserted to full depth at the start as in the Fisher and Cope method, but the tool skirt which covers the projecting end of the tube bears against the tube sheet as a lubricated sliding thrust. The ball bearing employed by the authors is not used.

The method of tube rolling, described in the paper, employs electric-motor drive and a relay adjusted after test upon samples of tubes. The thickness and hardness of samples control the current flow allowed for the rolling operation.

In Worthington practice, compressed-air motor drive is employed. Experiment likewise determines the correct tool setting. The limit of rolling is effected by means of a sleeve machined to the length determined by experiment upon the sample. This sleeve is placed over the mandrel between the tool body and the "travel limit" indicated in Fig. 6 of the paper and apparently exercises control of rolling as does the electric-drive and relay current-limit method.

In rolling with the method described in the paper, thicker tubes require more current. We thus assume the relay is reset for higher current flow. In Worthington practice, when overgage tubes are encountered, a longer stop sleeve is used to reduce the travel of the tapered mandrel.

In any case the tool should be entered to the full depth at the start of rolling as is done in both methods.

Authors' Closure

The familiar uniform-entrance method of rolling-in condenser tubes, described by J. F. Grace, produces joints in which the inside diameter of the tube is uniform when a stop sleeve of given length is used. This method was tried once by the authors' company, but the results were not as satisfactory as had been hoped for, since this method does not take into account the variations, permitted by commercial specifications, in the diameter of the tube-sheet holes and in the thickness of the tube wall.

In the production of joints in which the metal is uniformly yet not excessively cold-worked, and the axial push-out strength is reasonably uniform, this method using a stop sleeve can be applied only when all the tube-sheet holes are of identical diameter and the tubes have the same wall thickness and outside diameter.

Because such uniform dimensions are not found in commercial tubes and tube-sheet holes, the authors undertook to develop a technique of rolling tubes that would be independent of the common variations found. The paper describes the results obtained. The technique that they have developed might be applied to the air-driven expander, if a satisfactory control device were available. Such devices as can be obtained are not nearly as flexible as are those readily available for the control of the electric-motor-driven expander.

At the time of writing this closure, the authors are applying their improved technique to the rolling-in of about 2000 tube-hole plugs in two condensers that have been in service for several years. It is believed that this experience will be of sufficient interest to others so that it may be related in some detail.

About 1000 tubes are to be removed from each of the condensers serving two 50,000-kw turbogenerators in order to reduce the surface area. At one end, the joints between tube and tube sheet were sealed with packing held in place by threaded ferrules. The tube-sheet holes left open by the removal of those tubes were plugged by means of threaded plugs. At the other end, the joints were rolled into the tube sheet so that the simplest way to close these tube holes was to roll-in plugs made from short (5-in.) lengths of condenser tube, having one end sealed. The open end of each plug was prerolled to an outside diameter slightly larger than that of the average tube hole so that the plugs would fit snugly if not tightly in the holes, thus facilitating rolling-in.

In order to determine the correct setting of the current-limiting relay to be used in connection with the rolling-in of these plugs, detailed information which was not available, the authors conducted laboratory tests on a few joints using plugs rolled into a piece of brass tube sheet which was available. The only difference between the laboratory tests and the work on the condenser was in the tube-sheet holes. Those in the tube sheet used in the tests were freshly reamed to a fairly uniform size and the surfaces were smooth while those in the condenser tube sheet had not been rereamed after the tubes were removed, but had been cleaned by means of a wire brush. The values obtained in the test are given in Table 2, herewith.

TABLE 2 RESULTS OF LABORATORY TESTS IN ROLLING-IN 1-IN. PLUGS IN SAMPLE TUBE SHEETS

Specimen no.	1	2	3	4
Initial outside diameter of prerolled end of tube, in.	1.0140	1.0150	1.0155	1.0154
Initial inside diameter of prerolled end of tube, in.	0.9190	0.9190	0.9190	0.9190
Inside diameter of hole before rolling-in, in.	1.0138	1.0130	1.0145	1.0135
Inside diameter of end of tube after rolling but before pushing out, in.	0.923	0.9215	0.9245	0.9210
Elongation, in.	0.006	0.006	0.010	0.010
Push-out strength of joint, lb	1640	1430	2300	1550
Average Rockwell hardness E of unrolled tube				60
Average Rockwell hardness E of prerolled end of tube				84
Average Rockwell hardness E of rolled end of tube after pushing out				91

Measurements made on several tube holes, taken at random in the condenser tube sheet, revealed that the holes varied from 1.012 to 1.015 in. in diam. Such a variation would have no effect on the results of rolling-in tubes by the authors' method, and no importance was attached to the fact that this or a larger variation might exist. It is to be noted in this connection that, in the laboratory tests, the inside diameter of the rolled portion of the tubes, taken before pushing out the tubes, varied from 0.9210 in. (specimen no. 4) to 0.9245 in. (specimen no. 3) with an average value of 0.9225, and the axial load required to break the joints varied from 1430 to 2300 lb with an average value of 1730 lb.

The tests revealed that the motor-driven expander used in the tests, the same one that was to be used in rolling-in the plugs in the condenser, when operating on 115-v alternating current, drew 1.10 a, while running idle, and 2.35 a, maximum, while rolling-in a 1-in. plug to 0.010-in. elongation, the value that appeared to be the optimum. In consequence, the current-limiting relay was adjusted to operate at 2.35 a.

After this adjustment had been made, four additional plugs

[8] Design Engineer, Worthington Pump and Machinery Corporation, Harrison, N. J. Mem. A.S.M.E.

were rolled-in using the full automatic features of the technique. In addition, a dial indicator was attached to each plug while it was being rolled-in, in order to determine the elongation actually attained. The conditions which prevailed when rolling plugs into sample tube-sheet holes from which tubes had been removed duplicated almost exactly the conditions found in the condenser. After rolling-in, these plugs were pushed out by means of the testing machine in order to determine the push-out strength. It was found that, with the 2.35-a setting of the current-limiting relay, the plugs had been rolled to an average elongation of 0.009 in. and the push-out load varied from 1565 to 2000 lb, with an average value of 1760 lb per tube.

After the tests had been completed, which required about 1 day for one man, the expanding equipment used in the tests was taken to a condenser which was undergoing alteration and the authors' technique was applied in rolling-in the plugs. Since the tubes in the two condensers were all 1 in. OD and of approximately the same hardness, no resetting of the current-limiting relay was necessary when the tool was transferred from one condenser to the next as the work advanced, or for future work on the same size tubes.

The job of rolling-in the plugs in the condensers was assigned to a mechanic's helper, who had never operated a tube expander before. The elapsed time for rolling-in ten plugs in a row was noted. This was 90 sec, or 9 sec per joint. This time, of course, does not include the time necessary to insert the plugs into the tube-sheet holes and drive them into place.

Supplementary tests have been conducted in order to determine if satisfactory joints would be obtained when maximum and minimum commercial clearances prevailed and when tubes of various hardnesses were rolled according to the authors' improved technique.

In Table 3, each item represents the average results obtained on three or more individual joints.

Item 1 represents the minimum commercial tolerance in clearance and Item 2 the maximum.

Item 3 represents the softest tube that was readily available and Item 4 the hardest. In all of these cases, the cold working of the metal was not excessive in the authors' opinion, and the strength of the joints was ample and reasonably uniform, considering all the variables involved in the rolling of any joint. These data substantiate the belief previously expressed that this technique is universally applicable to the rolling of condenser tubes of commercial hardnesses and dimensional tolerances, and further, they leave no uncertainty regarding the suggested limitations of the technique voiced by Mr. Grace.

TABLE 3

Item	Tube diam, in.	Hole diam, in.	Hardness of tubes before rolling, Rockwell E	Elongation, in.	Push-out load, lb	Hardness of inside of rolled section, Rockwell E	Current to expander Idling, amp	Current to expander Max, amp
1	0.752	0.755	68	0.008	1728	90	0.82	1.25
2	0.748	0.760	67	0.008	1630	92	0.82	1.25
3	0.751	0.761	58	0.009	1670	88	0.82	1.25
4	0.750	0.760	77	0.007	1960	95	0.82	1.25

Heat Transfer and Fluid Resistances in Ljungstrom Regenerative-Type Air Preheaters

By HILMER KARLSSON[1] AND SVEN HOLM,[2] WELLSVILLE, N. Y.

This paper includes a review of the experiments and studies on the heat transfer and pressure drops in Ljungstrom regenerative-type air preheaters, as carried out some time ago by the Ljungstrom Steam Turbine Company in Sweden (1).[3] It also includes some theoretical investigations reported in the European technical press, as well as results of tests carried out at the Wellsville, N. Y., plant of the Air Preheater Corporation, on pressure drops across different types of heating surface. Partial results of the tests made in Sweden have been published in the European press and during the World Power Conference in Tokio in 1929 (2). Although this review covers particularly theories and tests underlying the principle of the Ljungstrom type of continuous regenerative air preheater, these theories can also be applied to other regenerative types of air preheaters. Regenerative air preheaters are either of the intermittent or the continuous type.

Nomenclature

The following nomenclature is used in this paper:

a = height of notch (heating surface elements, see Fig. 3), mm
C = conductivity, Btu per sq ft per deg F per hr, 1 ft thick
c_1 = specific heat of gas at constant pressure, Btu per lb per deg F
c_2 = specific heat of air at constant pressure, Btu per lb per deg F
c_3 = specific heat of steel in heating elements, Btu per lb per deg F
$d = 2a + 0.7\,u$ = hydraulic diameter referred to Ljungstrom type heating surface, mm
E_1 = sensible heat of gas flowing $G_1 \times c_1$, Btu per deg F per hr
E_2 = sensible heat of air flowing $G_2 \times c_2$, Btu per deg F per hr
E_3 = sensible heat in heating elements $W \times c_3$, Btu per deg F
e = base of natural logarithms = 2.718
F = total effective heating surface = $F_1 + F_2$, sq ft
F_1 = heating surface gas side = $r \times F$, sq ft
F_2 = heating surface air side $(1 - r) \times F$, sq ft
G_1 = weight of gas flowing, lb per hr
G_2 = weight of air flowing, lb per hr
k_1 = heat-transfer coefficient of gas to heating elements, Btu per sq ft per deg F per hr
k_2 = heat-transfer coefficient, heating elements to air, Btu per sq ft per deg F per hr
L = length of path of heat flow, ft
L = depth of heating surface, ft
m = ratio of sensible heats, air to gas $= \dfrac{G_2c_2}{G_1c_1}$
N = revolutions per minute, rpm
n = revolutions per hour, rph
P = perimeter of cross section, ft or in.
ΔP = pressure drop, in. wg
Q = quantity of heat, Btu per hr
r = fraction of effective heating surface in gas stream
S = area of cross section, sq ft or sq in.
s = plate thickness, in.
ta = mean temperature of air leaving, F
ta_1 = air temperature entering, F
$t'a_2$ = air temperature leaving at beginning of air period, F
$t''a_2$ = air temperature leaving at end of air period, F
tg = mean temperature of gas leaving, F
tg_1 = gas temperature entering, F
$t'g_2$ = gas temperature leaving at beginning of gas period, F
$t''g_2$ = gas temperature leaving at end of gas period, F
t'_{f1} = temperature of heating elements at point of gas entering and at beginning of gas period, F
t''_{f1} = temperature of heating elements at point of gas entering and at end of gas period, F
t'_{f2} = temperature of heating elements at point of gas leaving and at beginning of gas period, F
t''_{f2} = temperature of heating elements at point of gas leaving and at end of gas period, F
$\Delta t'$ and $\Delta t''$ = temperature difference, F
Δt_m = mean temperature difference, F
u = depth of undulation (heating-surface elements; see Fig. 3), mm
u/d = specific undulation
U = over-all heat-transfer rate, Btu per sq ft per deg F per hr
V = velocity, fps
V_ρ = mass velocity in free duct area, lb per sq ft per sec
$V_0\rho$ = mass velocity through element channels, lb per sq ft per sec
W = weight of heating elements, lb
X = direction of gas flow
Y = temperature scale
Z = direction of rotation
β = correction factor for heating surface
η_a = air temperature efficiency $= \dfrac{ta - ta_1}{tg_1 - ta_1}$
η_g = gas temperature efficiency $= \dfrac{tg_1 - tg}{tg_1 - ta_1}$
η = mean temperature efficiency $= \dfrac{\eta_a + \eta_g}{2}$
η' = mean temperature efficiency when rotor speed is taken into account

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[1] Chief Engineer, Air Preheater Corporation. Mem. A.S.M.E.
[2] Research Engineer, Air Preheater Corporaton.
[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Heat Transfer Division and presented at the Semi-Annual Meeting, Cleveland, Ohio, June 8–10, 1942, of The American Society of Mechanical Engineers.

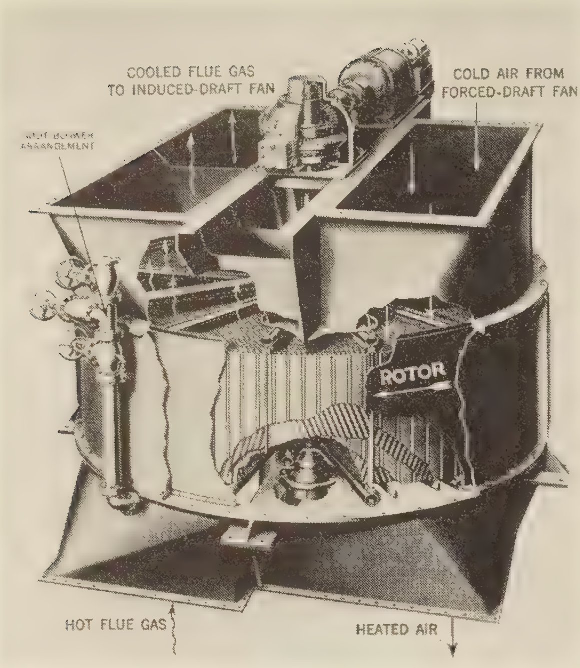

FIG. 1 VERTICAL-FLOW TYPE OF LJUNGSTROM AIR PREHEATER

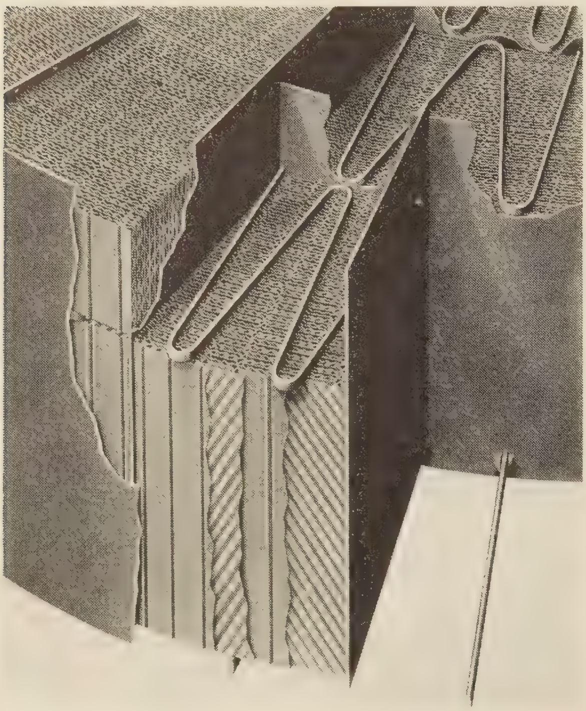

FIG. 2 ENLARGED CUTAWAY DETAIL OF NOTCHED UNDULATED TYPE HEATING SURFACE

η^* = mean temperature efficiency when heat conductivity in gas-flow direction of heating surface is not taken into account

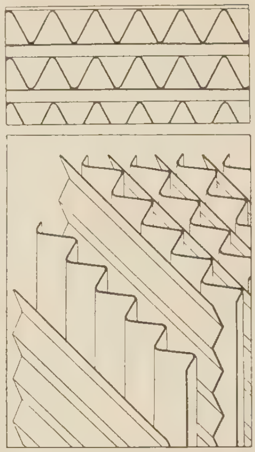

FIG. 4 CORRUGATED UNDULATED TYPE HEATING SURFACE

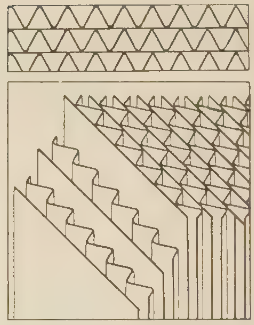

FIG. 5 CORRUGATED PLAIN TYPE HEATING SURFACE

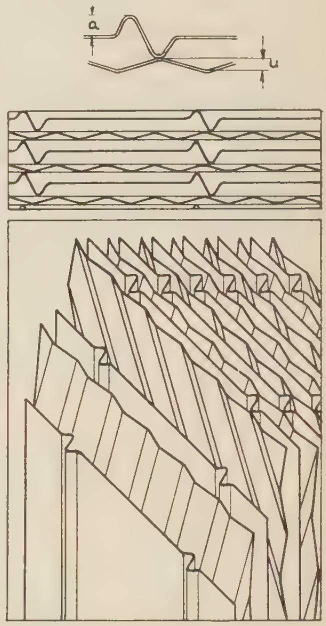

FIG. 3 HEATING SURFACE SHEETS WITH INCLINED UNDULATIONS SET BETWEEN VERTICALLY NOTCHED SHEETS

$\Delta\eta$ = drop in temperature efficiency due to conduction of heat in elements in direction of gas flow
ρ = density, lb per cu ft
τ_1 = time of rotating heating surface in gas stream, hr
τ_2 = time of rotating heating surface in air stream, hr

DESCRIPTION OF CONTINUOUS REGENERATIVE HEATER

The continuous regenerative principle employed in the Ljungstrom air preheater includes the use of a slowly moving rotor containing the heating surface. Each revolution produces a complete cycle of exchange in which heat from the hot gases is constantly absorbed by the heating surface in the rotor and given up as the rotation moves it into the path of the combustion air.

Fig. 1 is a view of the vertical-flow type of Ljungstrom air preheater with part of the housing cut away to show the integral parts. Fig. 2 is an enlarged detail of the rotor with part of the shell cut away to show the arrangement of present standard-type heating surface known as the notched undulated. As shown in this illustration the heating surface is arranged in two layers. Each layer of heating surface is designed to suit the particular operating conditions to be encountered. This type of heating surface is most widely used at present in the Ljungstrom type of air preheater and is illustrated in Fig. 3. Fig. 4 illustrates the cor-

rugated undulated type of heating surface used prior to the adoption of the notched undulated type, while Fig. 5 illustrates the corrugated plain type of surface which was used at the time this type of air preheater was first made commercially available. Figs. 4 and 5 are included to illustrate the development and also the great variation in types of surface that can be used and will be referred to later.

Efficiency of Ljungstrom Continuous Regenerative Type Air Preheater

Throughout this paper, the efficiency used to express the heat recovery of the air preheater is the temperature efficiency. By actual measurement of temperatures of gas and air entering and leaving, this efficiency can be calculated as follows:

$$\text{Gas temperature efficiency } \eta_g = \frac{tg_1 - tg}{tg_1 - ta_1}$$

$$\text{Air temperature efficiency } \eta_a = \frac{ta - ta_1}{tg_1 - ta_1}$$

$$\text{Mean temperature efficiency } \eta = \frac{\eta_g + \eta_a}{2}$$

In these formulas, tg is the theoretical temperature of the gases leaving the air preheater that would be obtained if no air infiltration toward the gas side of the unit takes place.

The temperature efficiency varies with the proportion of areas for gas and air flow through the heater and also with the ratio of sensible heats in air and gas flowing. A complete theoretical treatise covering the influence of these factors has been carried out by Messrs. Hakanson and H. Zander (3). Equation [1] and

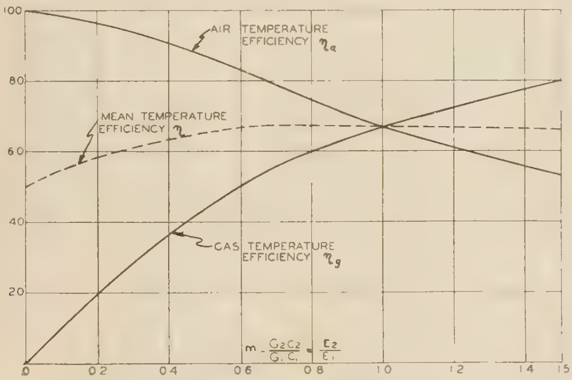

FIG. 6 TEMPERATURE EFFICIENCES

the diagram in Fig. 6 are taken from this treatise (3) and show the final results of the investigation

$$\eta_g = \frac{1 - e^{\frac{k_1F}{G_1c_1}\left(\frac{1}{m}-1\right)\frac{0.525\,(1-r)^{0.2}\times m^{0.8}}{1+0.81\left(\frac{1-r}{r}\right)^{0.2}\times m^{0.8}}}}{1 - \frac{1}{m}\, e^{\frac{k_1F}{G_1c_1}\left(\frac{1}{m}-1\right)\frac{0.525\,(1-r)^{0.2}\times m^{0.8}}{1+0.81\left(\frac{1-r}{r}\right)^{0.2}\times m^{0.8}}}} \quad \ldots\ldots [1]$$

From Equation [1], it can be seen that η_g is a function of the heat-transmission coefficient, effective heating surface, and sensible heat in gas flowing. When, from measurements, η_g has been determined, k_1 can be calculated by using Equation [1]; k_2 can also be calculated by using Equation [1], using values corresponding to the air side of the preheater. In the special case, when $m = 1$, the use of Equation [1] will result in the indeterminate form $\eta = 0/0$. By differentiation with respect to m, the equation becomes

$$\eta_g = \frac{\dfrac{k_1F}{G_1c_1} \times \dfrac{0.525(1-r)^{0.2}}{1+0.81\left(\dfrac{1-r}{r}\right)^{0.2}}}{1 + \dfrac{k_1F}{G_1c_1} \times \dfrac{0.525(1-r)^{0.2}}{1+0.81\left(\dfrac{1-r}{r}\right)^{0.2}}} \quad \ldots\ldots [2]$$

Fig. 6 shows how η, η_g, and η_a vary with m for a given set of conditions.

Effect of Air Leakage

In establishing the actual efficiency of an air preheater on the basis of test data on actual installations, the question of air leakage must be considered.

From quantitative analysis of gases entering and leaving an air preheater, the increase in gas quantity caused by air infiltration is determined. Knowing this and the temperature of the air entering the air preheater, the reduction in gas exit temperature caused by infiltration of cold air is estimated, thus arriving at the gas outlet temperature used to determine the efficiency of the air preheater.

Influence of Ratio of Sensible Heats—Air to Gas Through Preheater

The influence of this ratio (1) is shown in Fig. 7. According to this diagram, this relation has no practical influence in normal cases but at very high mean temperature efficiencies and low ratios between sensible heats in air and gas, the effect is of importance.

Example: Take the mean temperature efficiency of 70 per cent, and $m = 1$, and $m = 0.6$, we then obtain, according to the diagram

$$\frac{G_1c_1 \times G_2c_2}{G_1c_1 + G_2c_2} \times \frac{1}{UF} = 0.215 \text{ and } 0.186\text{, respectively.}$$

In other words, an increase in effective total heating surface of about 15 per cent is required at the lower m value for the same

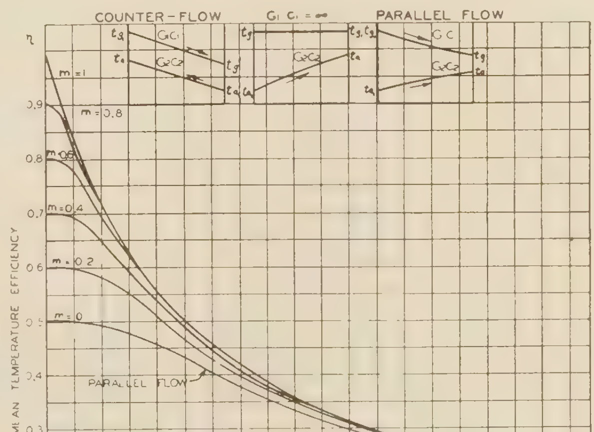

FIG. 7 MEAN TEMPERATURE EFFICIENCY FOR PARALLEL FLOW AND COUNTERFLOW; $m = (G_2c_2/G_1c_1)$

temperature efficiency if the sum of gas and air flow in both cases is equal.

Influence of Rotor Speed on Temperature Efficiency

A very complete study as to the influence of the rotor speed on the efficiency of a Ljungstrom continuous regenerative type of air preheater has been made by Gustav Boestad of the Ljungstrom Steam Turbine Company in Stockholm, Sweden, the results of which have been published (4). The following note is abstracted from this paper:

As the rotor revolves, the temperature of the heating surface increases while passing through the gas compartment so that the temperature is higher at the end of that period than at the beginning. The rate of increase in temperature is higher at the beginning than at the end of the period because of the higher temperature difference existing at the beginning. By the same token, the temperature of the heating surface decreases at a higher rate in the beginning of the cooling period in passing through the air compartment.

Fig. 8 shows a diagram of the temperature changes mentioned. In this illustration tg_1 indicates the constant inlet gas temperature. The solid lines t'_{f1}-t''_{f1} and t'_{f2}-t''_{f2} denote the temperature at the hot and cold ends of the heating surface during the heating period. The dotted lines similarly designated refer to the cooling part of the cycle or air period.

The lines $t'a_2$-$t''a_2$ and $t'g_2$-$t''g_2$ show the temperature variation of the air and gas leaving the air preheater as affected by rotor travel.

A complete mathematical analysis of these curves is made by Boestad (4).

The space diagram, Fig. 9, gives a comprehensive idea of the temperature cycles in a Ljungstrom type of air preheater. The variation in efficiency with variable rotor speeds can be estimated from Equation [3], which can be used for variations in efficiency greater than 0.05 and when the temperature efficiency is greater than 0.5.

$$\eta - \eta' = \frac{(\eta)^2}{10} \times \left(\frac{E_1}{E_3 n}\right)^2 \qquad [3]$$

This formula is not true for $n = 0$, because then $\eta - \eta'$ becomes infinite.

For very low speeds, all of the regenerative mass (heating surface) will reach the temperature of the gases when passing through the gas compartment and reach the temperature of the cold air when passing through the air compartment. The efficiency will then depend, for a limiting condition, when $n = 0$ only, upon the storage capacity. We then obtain

$$\eta = \frac{Q}{Q_{\max}} = \frac{nE_3(tg_1 - ta_1)}{E_1(tg_1 - ta_1)} = \frac{nE_3}{E_1} \qquad [4]$$

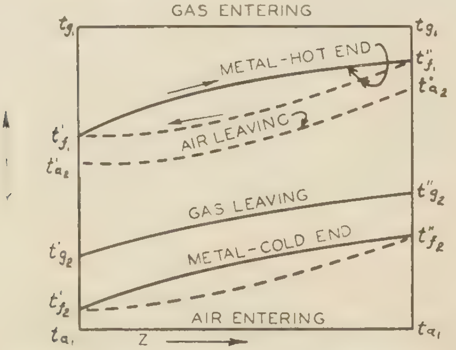

Fig. 8 Temperatures in Preheater as Affected by Rotor Travel

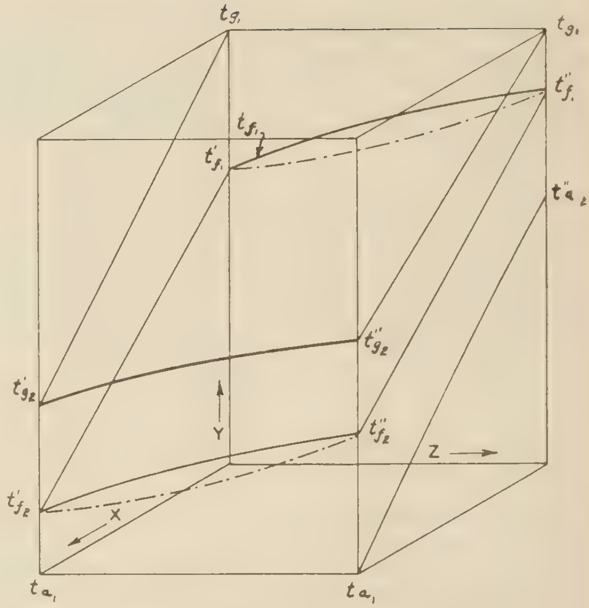

Fig. 9 Space Diagram of Temperatures in Preheater
(Symbols are the same as in Fig. 8.)

the slope of the temperature-efficiency curve, as depending upon n, for $n = 0$ is then by differentiating

$$\frac{d\eta}{dn} = \frac{E_3}{E_1}$$

The following example will explain the use of Equation [3]:

Gas quantity	=	220,500 lb per hr
Sensible heat in gases E_1	=	61,800-Btu per deg F per hr
k_1	=	5.12 Btu per sq ft per deg F per hr
k_2	=	7.17 Btu per sq ft per deg F per hr
F_1	=	50,400 sq ft
F_2	=	34,500 sq ft
η	=	0.69 at 180 rph
E_3	=	4500 Btu per deg F

For $n = 10$

$$\eta - \eta' = \frac{0.69^2}{10} \times \left(\frac{61,800}{4500 \times 10}\right)^2 \cong 0.09$$

$$\eta' = 0.69 - 0.09 = 0.60$$

The slope of the efficiency curve at $n = 0$ is

$$\frac{d\eta}{dn} = \frac{4500}{61,800} = 0.073$$

Fig. 10 shows a decrease in temperature efficiency with a decrease in rotor speed. It is evident from the curve that above a speed of 30 rph the gain in efficiency possible by increase in rotor speed can be neglected.

Temperature of Heating Surface

It is important that every portion of the heating surface in an air preheater be maintained at a temperature above the dew point of the gases in order to avoid difficulties with corrosion. This requires that the designer should have a knowledge of the actual metal temperatures to be anticipated, besides which the following rules should be observed:

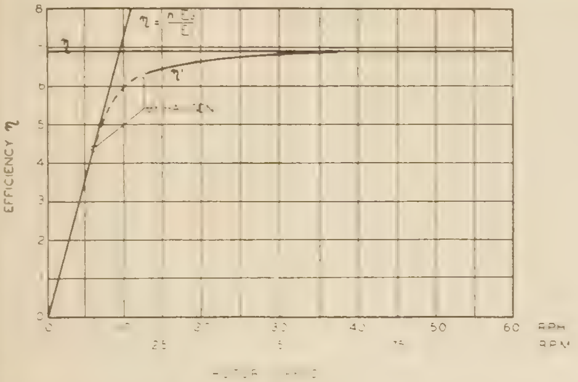

Fig. 10 Influence of Rotor Speed on Air-Preheater Efficiency

1 The use of high gas-mass velocity and low air-mass velocity will maintain the highest possible temperature of the heating surface.

2 Avoid turns, whirls, and dead corners with resulting possible undercooling of the heating surface.

3 Avoid pulsating admission mainly on the cold-air inlet.

4 Provide accessibility to the heating surface, as well as practical and efficient cleaning facilities.

The heating surface of a regenerative-type air preheater passes through periodical temperature variations, about the character of which most of us have the wrong conception.

Fig. 11 from reference (5) shows a typical method of determining the minimum temperature of the heating surface. In this illustration, the necessary data used are from the previous example under the heading "Influence of Rotor Speeds on Temperature Efficiency."

For the recuperative type of air preheater, the heat-transmission coefficient on the gas side is laid off at the inlet-air temperature of 80 F to the left, and the heat-transmission coefficient on the air side is laid out to the same scale at the gas-outlet temperature to the right. The line connecting these two points intersects the temperature scale at about 141 F.

For the regenerative type of air preheater, the heat-transmission coefficient on the gas side multiplied by the proportionate time the heating surface is in the gas stream is laid off to the left at the inlet-air temperature; and the heat-transmission coefficient on the air side multiplied by the proportionate time the heating surface is exposed to the air stream is laid off to the right to the same scale as the outlet-gas temperature. The line connecting these points intersects the temperature scale at about 153 F. The diagrams, Fig. 11, are based upon the assumption that fluid temperatures and mass velocities are uniform at the inlet to the heating surface. They show that, for similar conditions, the metal temperature in a Ljungstrom type of heater is higher than that in the recuperative type heater.

Fig. 12 illustrates the fluctuation in the temperature of the heating surface with the rotor speed. The data used in plotting this diagram are the same as those used for the diagram shown in Fig. 11 and in the previous example. If, because of cooling the heating surface below the dew point, corrosion occurs, it is desirable to provide that the portion of the surface so affected will be readily replaceable. Fig. 2 illustrates clearly the provision made in the design of the Ljungstrom air preheater to accomplish this purpose. The heating surface is arranged in two layers. The depth of each layer is proportioned so that corrosion under the worst anticipated conditions will occur in the shallow layer only, which can be replaced at low cost. In this connection, it will be noted that corrosion of the heating surface in a regenerative type of air preheater does not reduce its efficiency until there is an actual loss in surface area. This condition, however, will not cause mixing of the two fluids.

Fig. 13 shows the effect of travel through gas and air compartments on the temperature of the metal of regenerative type air preheaters of the Ljungstrom type at 180 rph. The space between gas and air and vice versa indicates the sealing period.

Influence of Heat Conduction of Heating Surface in Direction of Gas Flow (1)

The influence of this heat flow is shown by the diagram in Fig. 14.

Example: Let it be assumed that the sensible-heat content of air and gas are equal, $-E_1 = E_2$; that the heat-transmission coefficient on the gas and air sides is equal, $k_1 = k_2 = k$; and that the

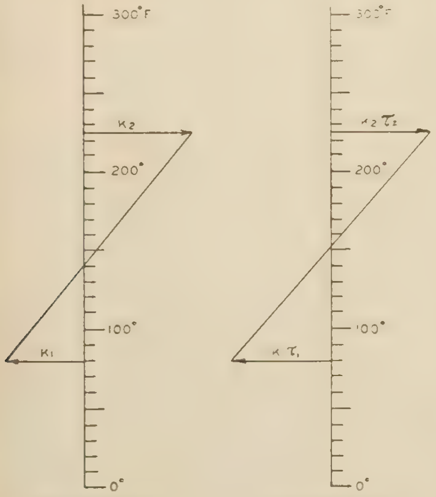

Fig. 11 Graphical Method of Determining Minimum Temperature of Heating Surface When Heat-Transfer Rates K and Time τ Are Known

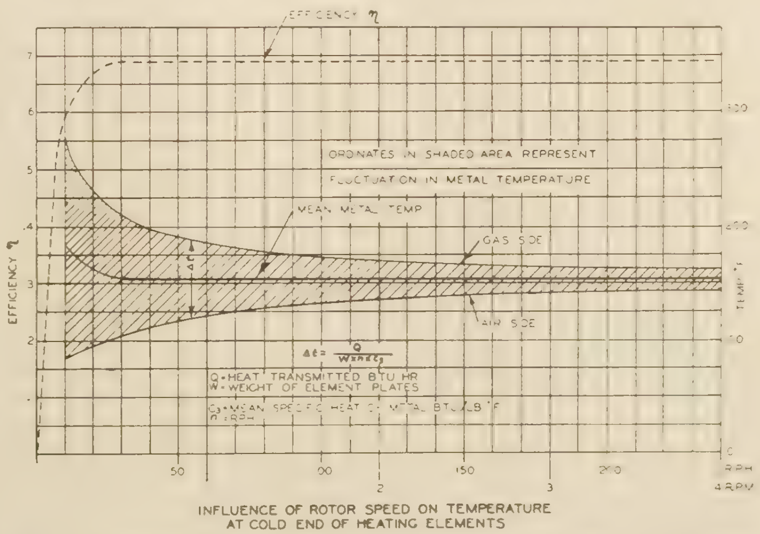

Fig. 12 Influence of Rotor Speed on Temperature at Cold End of Heating Elements

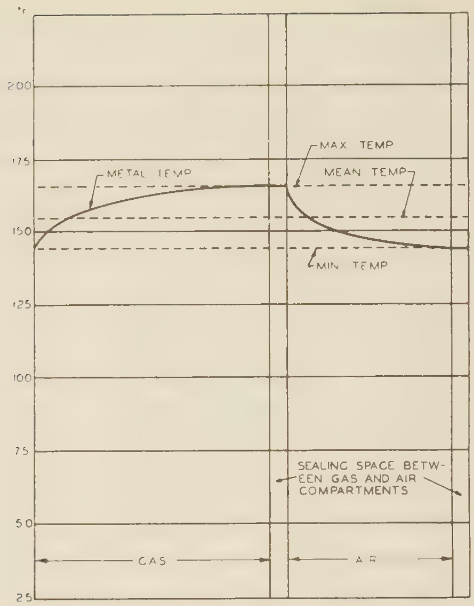

Fig. 13 Temperature Variation at Cold End in Regenerators During One Gas and Air Period at 3 Rpm

plate-temperature difference between inlet and outlet reduced through the heat conductivity of the plate from $\Delta t'$ to $\Delta t''$. Then the heating surface must be increased in the relation

$$\frac{1}{\beta} = \frac{1}{2\eta} \times \frac{1-\eta}{1-\dfrac{\Delta t''}{\Delta t}} \times \log_e \frac{1-\eta \times \dfrac{\Delta t''}{\Delta t'}}{1+\eta \times \dfrac{\Delta t''}{\Delta t'} - 2\eta} \quad \ldots\ldots [6]$$

$$\eta = \frac{1}{1+\dfrac{E_1}{\beta UF}}; \quad \Delta\eta = \eta'' - \eta; \quad \eta'' = \frac{1}{1+\dfrac{E_1}{UF}}$$

the maximum temperature efficiency is

$$\eta_{max} = \frac{1}{2-\dfrac{\Delta t''}{\Delta t'}} \quad \ldots\ldots [7]$$

$$\frac{\Delta t''}{\Delta t'} = 1 - \frac{2}{L}\sqrt{\frac{C \times s \times {}^{1}/_{12}}{2k}} \quad \ldots\ldots [8]$$

For the 30-deg notched-undulated elements with a hydraulic diameter of d = 7 mm, a mass velocity of gas of 0.82 lb per sq ft per sec, a depth of element in the direction of travel of flow = $23^1/_2$ in., and an element thickness of No. 24 USG, the decrease in mean temperature efficiency is less than 0.2 per cent. It increases slowly with increasing plate thickness and decreasing hydraulic diameter and mass velocity of fluids. At the especially low mass velocity of 0.2 lb per sq ft per sec and d = 7 mm, it has a value of 0.6 per cent.

Various Types of Heating Surface

One of the main advantages of the continuous regenerative type of air preheater is its adaptability to the use of heating surfaces of different shapes and forms, while in the case of the recuperative air preheater, the designer is more confined to the conventional

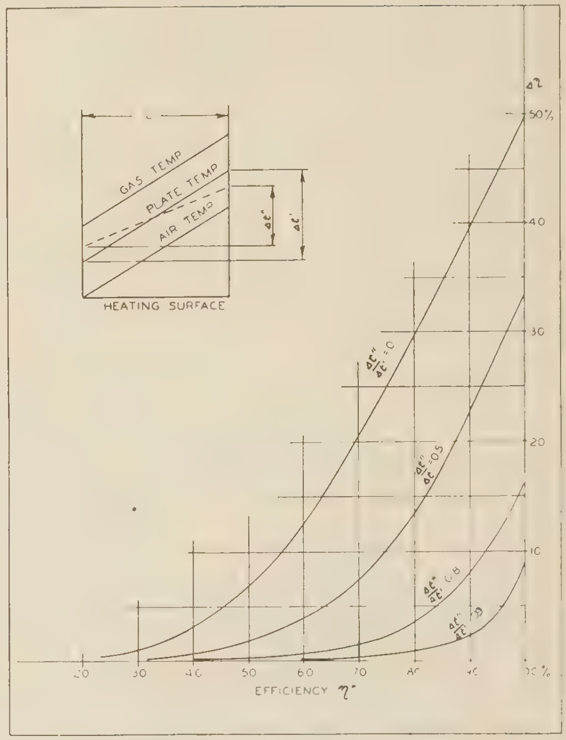

Fig. 14 Decrease in Mean Temperature Efficiency $\Delta\eta$ Due to Heat Conductivity of Plate in Gas-Flow Direction

forms of heating surface available. In order to determine the type of heating surface most suitable to the requirements to be met with an air preheater of the regenerative type, considerable experimental work was required.

In the development of the Ljungstrom air preheater, a great many different forms of heating surface have been tried out, three of which have been commercially used. The type of surface first used commercially in the Ljungstrom air preheater was that illustrated in Fig. 5 and termed the corrugated plain type.

Further experimentation led to the adoption of the corrugated undulated type of heating surfaces, shown in Fig. 4. This second commercial type of heating surface adopted had the advantage of providing fewer contacts between the individual sheets, thus resulting in a higher percentage of effective surface.

Further laboratory studies led to the adaptation of the notched undulated type of surface now used as standard in the Ljungstrom regenerative type of air preheater, which type of surface is illustrated in Figs. 2 and 3, the form of which has been previously described.

Prior to the adoption of this type of surface in general practice in the United States, two size 18 Ljungstrom air preheaters, installed at the Kearny, N. J., plant of the Public Service Electric and Gas Company, were equipped with the notched undulated type of surface. At this plant, a total of six size 18 vertical-flow-type Ljungstrom units are installed in connection with three 2350-hp boilers, the air preheaters being arranged in pairs on each boiler. Elaborate boiler tests were perfomed on the boiler having the Ljungstrom air preheaters equipped with the notched undulated type of heating surface and also on one of the boilers having its air preheaters equipped with the corrugated undulated type of heating surface. These tests gave the following results:

1 That the mean temperature efficiency of the air preheater was increased 2.2 per cent.

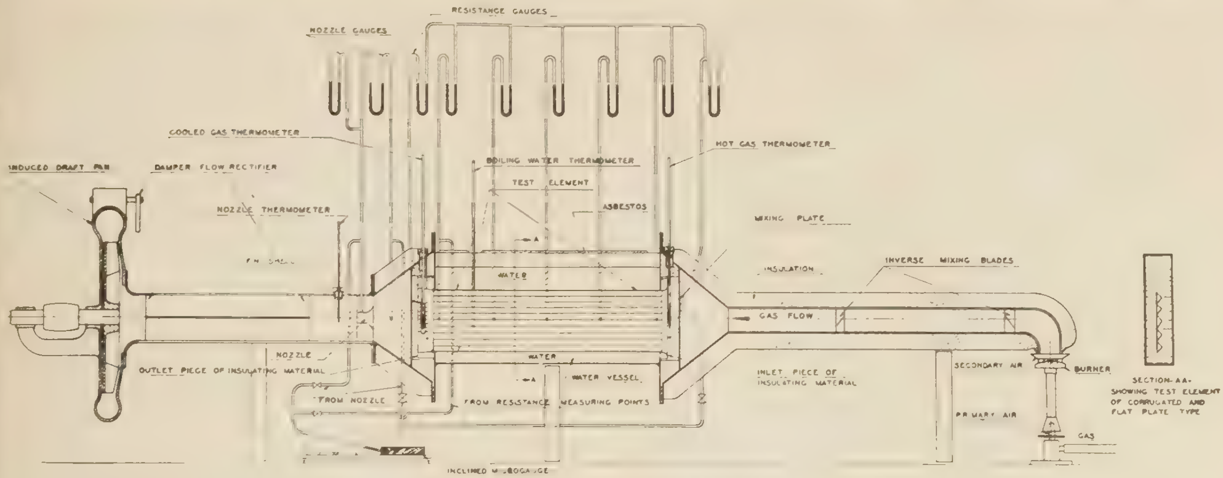

FIG. 15 SCHEMATIC ARRANGEMENT OF TESTING APPARATUS

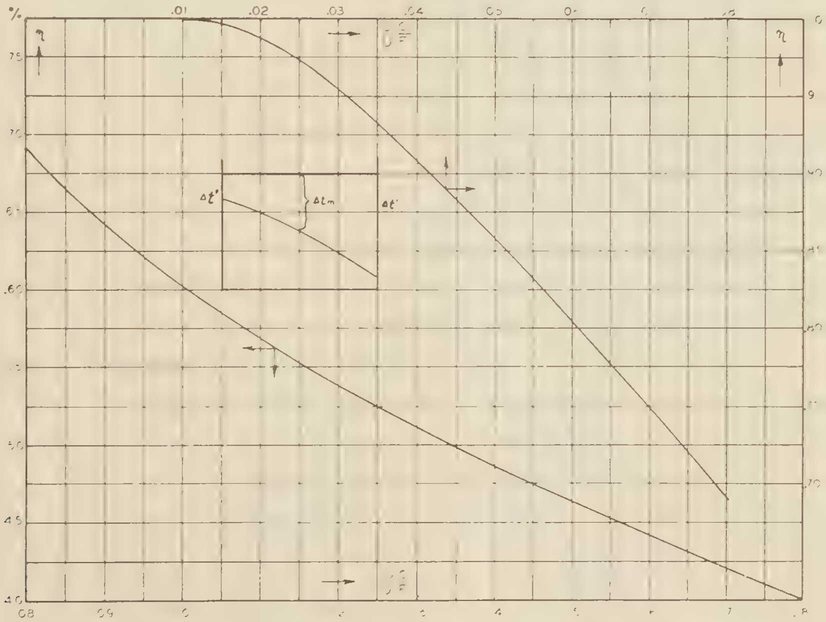

FIG. 16 TEMPERATURE EFFICIENCY WITH CONSTANT TEMPERATURE ON ONE MEDIUM

2 That the friction losses on the gas side of the air preheater decreased 45 per cent.

3 That the friction losses on the air side of the air preheater decreased 40 per cent.

In this connection, it is also interesting to note that, with the notched undulated type, 24 per cent more heating surface was provided in the same rotor space than was possible with the corrugated undulated type of surface, and that, in spite of the large increase of heating surface, the friction losses were substantially decreased. These tests, which were performed with the greatest possible care so as to obtain comparative results, substantiated the results of the laboratory work carried out by the Ljungstrom Steam Turbine Company in Sweden, which work will now be described.

Laboratory Tests on Heating Surface

Fig. 15 shows a schematic arrangement of the testing apparatus used by the engineers of the Ljungstrom Company in investigating heat-transfer rates and friction losses for different types of heating surface.

Heated air was produced in a gas burner from which the gases had to pass through a pipe provided with mixing blades to assure a uniform temperature. To the end of this pipe was bolted a flat pipe built up of the elements to be tested. The gas passed inside of this flat pipe and was cooled down by boiling water in the surrounding vessel. By this method the temperature, on the outside of the heating surface being tested, was kept constant, since the heat given up by the gas was sufficient to maintain the water at the boiling point. At very low loads, however, it was necessary to provide heating of the water vessel to keep the water at the boiling temperature. After passing the pipe, the cooled gas was passed through a chamber provided with a calibrated nozzle by which the flow was measured.

Calibrated thermometers for measuring the temperature of the gas entering and leaving the elements were placed immediately before and after the test section. A thermometer in the gas flow

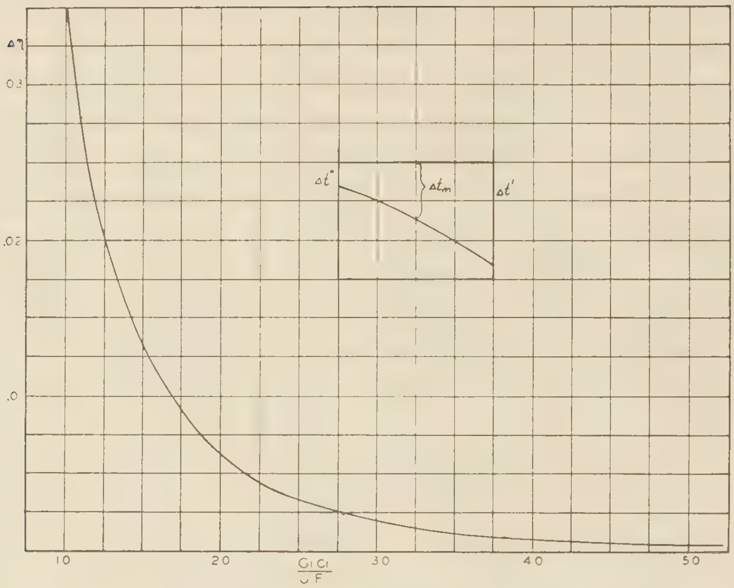

Fig. 17 Temperature Efficiency With Constant Temperature on One Medium

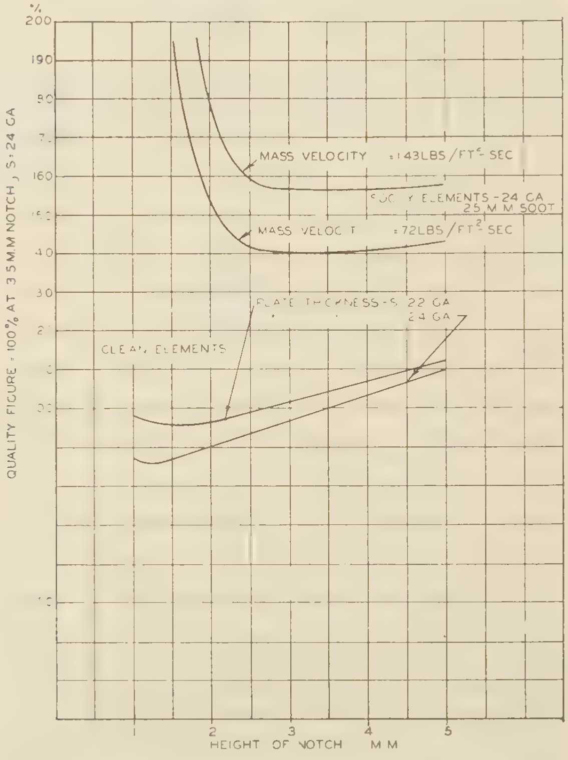

Fig. 18 Influence of Soot and Plate Thickness on Quality Figure Q for 30-Deg Notched-Undulated Elements

from the nozzle gave the temperature at this point. The temperature of the water was also observed by means of a calibrated thermometer.

The pressure drops in the nozzle were read on a U-gage for the higher velocities and on an inclined microgage for the lower ones.

The accuracy in measuring the outlet-gas temperature was improved by means of a mixing chamber filled with small glass balls between the outlet and the thermometer and also by enclosing the thermometer-fin pocket in the outlet-gas flow in such a way that no gas could pass by on the sides.

With a small apparatus like this, it is very important that no leakage take place. Therefore, before each test, the inlet and outlet were closed and the entire apparatus placed under slight internal pressure in order to check for possible leaks.

For each of the tests made, the mass flow and the temperature efficiency of the various elements tested was calculated. Then, from the formula $\eta = 1 - e^{-\frac{UF}{Gc}}$ the heat-transmission coefficient U was calculated. In this formula Gc is the sensible heat of the gas flowing over the surface, and F is the total effective heating surface, expressed in square feet.

Because of the slight resistance to heat flow on the water side of the elements, the heat-transmission coefficient on the gas side k_1 was considered equal to the total heat-transmission coefficient U. Figs. 16 and 17 show the curves from which U was calculated from the results of the experiments. In these diagrams

$$\eta = \frac{\Delta t' - \Delta t''}{\Delta t'} = 1 - e^{-\frac{UF}{Gc}} = \eta_{\text{approx}} - \Delta\eta \qquad [10]$$

$$\frac{\Delta t_m}{\Delta t'} = \eta \times \frac{Gc}{UF} \qquad [11]$$

$$\eta_{\text{approx}} = \frac{1}{0.5 + \frac{Gc}{UF}} \qquad [12]$$

Complete details of the foregoing test were given in a paper by Messrs. Alf Lysholm and E. Edenholm at the World Power Conference in Tokio in 1929 (2).

Influence of Height of Notch, Plate Thickness, and Soot

The effects of all of these variables are illustrated in Fig. 18. The ordinates in this diagram represent a figure of quality

$$q = \text{const} \times \frac{F}{G} \times \Delta p \qquad [13]$$

For purposes of comparison, an element employing 3.5 mm depth of notch and a plate thickness of No. 24 USG is regarded as unity or 100 per cent. It will be seen that increasing the height of the notch increases the figure of quality, which means that the heating surface must be increased in proportion. From the two upper curves on this diagram, it will be noted that the figure of quality rises very sharply for elements of shallow notches when carrying a thin layer of deposit, such as soot.

Influence of Soot on Pressure Drop and Heat Transfer for 30-Deg Notched-Undulated Elements (1)

The effect of this is shown in Fig. 19. As would be expected, the pressure drop increases with increased thickness of soot deposits, but the heat-transmission coefficient also increases at low mass velocity and decreased depth of notches. This indicates one of the noteworthy differences between recuperative and regenerative type air preheaters, as a deposit of soot and fly ash will retard the heat transfer in a recuperative type of unit, while in a regenerative type of unit, it has a tendency to aid the heat-transfer rate because of the fact that the deposit on the surface in

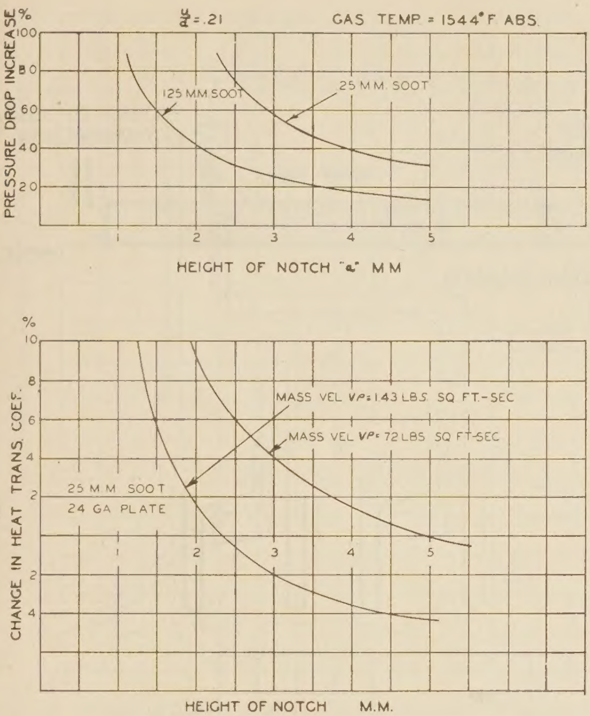

Fig. 19 Influence of Soot on Pressure Drop and Heat-Transfer Coefficient for 30-Deg Notched and Undulated Elements

the case of the regenerative type of air preheater acts as a heat-storage substance, as well as the metal in the heating surface itself.

However, excessive deposits on the heating surface obviously increase the fluid resistance with the result that if the deposit is left to build up without being checked, the unit becomes inoperative and this is true in all types of units.

Effect of Hydraulic Diameters[4] on Heating-Surface Requirements and Headroom Required for Air Preheaters

At first thought, it would seem that differences in the principle of operation between recuperative and regenerative air preheaters should not cause any great difference in the size. In both types, the heat transfer follows the same general principles. Both the heat conductance in the recuperative type and the heat storage in the heating-surface elements of the regenerative type can be neglected, when compared to the heat transfer from the gas to the heating surface, and from the heating surface to the air. However, by studying Fig. 20, an important influencing factor is quite apparent. This diagram shows the requirements of heating surface and headroom for preheaters of equal capacity, plotted as a function of the hydraulic diameters of the gas and air passages, allowing equal friction losses for all types of preheaters. It can be seen that the use of a small hydraulic diameter is a most effective means of increasing rates of heat transfer and consequently reduces headroom requirements.

For regenerative types of air preheaters, where the gas stream flows parallel to, but countercurrent to the air stream, it is possible to utilize fully the advantage of a small hydraulic diameter and to design the heating surface entirely from the standpoint of manufacture, capacity, and freedom from clogging.

[4] Diameter $d = 4S/P$ for cross sections of flow generally. Diameter $d = 2a + 0.7u$ for cross sections of flow in channels formed by Ljungstrom type heating surfaces.

Results of Pressure-Drop Experiments at Wellsville, N. Y.

The purpose of the experiments, carried out by the authors' company at Wellsville, N. Y., was to determine the fluid resistance through the heating surface with channels of different hydraulic diameters and to determine the effect of different types of seals on the fluid resistance. The apparatus used consisted of a test tunnel as illustrated in Fig. 21, having a square section where the elements or heating surface to be tested was placed.

Measuring stations were provided in the test section for a nine-point static-pressure traverse before and after the surface to be tested. The square test section was extended into a round section with the same cross-sectional area as the test section. Two Pitot tubes were located in the round section at a distance of $7^1/_2$ diam, each arranged for a ten-point traverse to determine the air velocity. Different diameter orifices were used to vary the air flow and thus obtain readings for different mass velocities. The temperature of the air passing over the surface was taken by calibrated thermometers at the orifice.

Each of the two Pitot tubes was connected to a set of gages, one water-filled U-gage for total pressure, one Ellison inclined vertical gage for static pressure, and one Ellison differential gage for velocity-pressure measurements. Each set of gages was mounted on a panel. For measuring static-pressure drops in the test section, an inclined vertical gage was used. For taking the static-pressure impulses in the test section, a special static-pressure tip was developed during these tests to assure accurate results.

Fig. 22 shows the results for a heating surface made up from sheets with 3.25-mm notches alternated with sheets having 1.725-mm undulations with the heating surface in two layers of $23^1/_2$ in. and $8^1/_2$ in. depth and a $^1/_2$-in. spacer between the two layers. Curve B shows the pressure drop without seals and curve A the pressure drop with two bulb-type seals located at the inlet and two at the outlet of the test section. The ratio of projected area

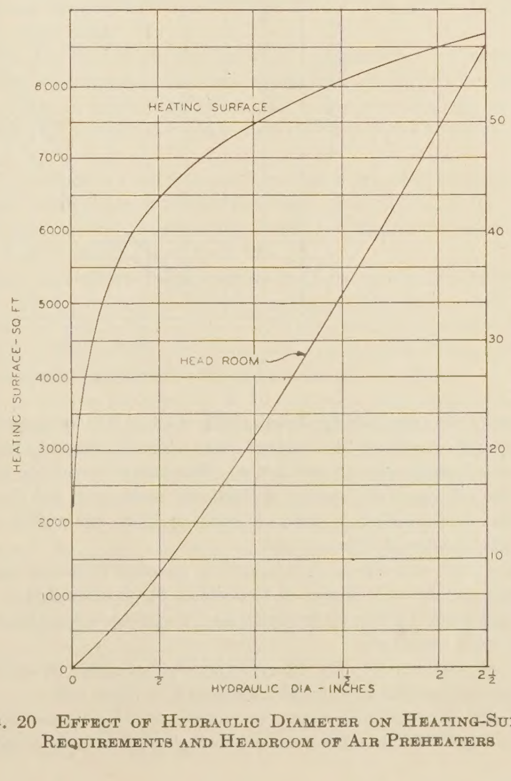

Fig. 20 Effect of Hydraulic Diameter on Heating-Surface Requirements and Headroom of Air Preheaters

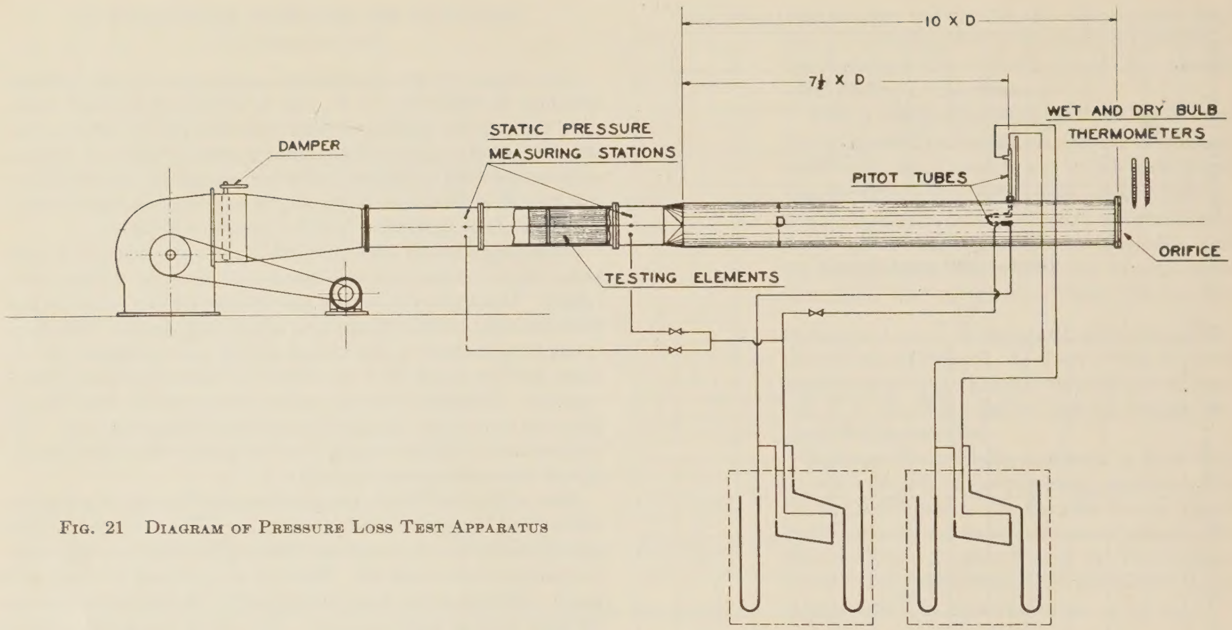

Fig. 21 Diagram of Pressure Loss Test Apparatus

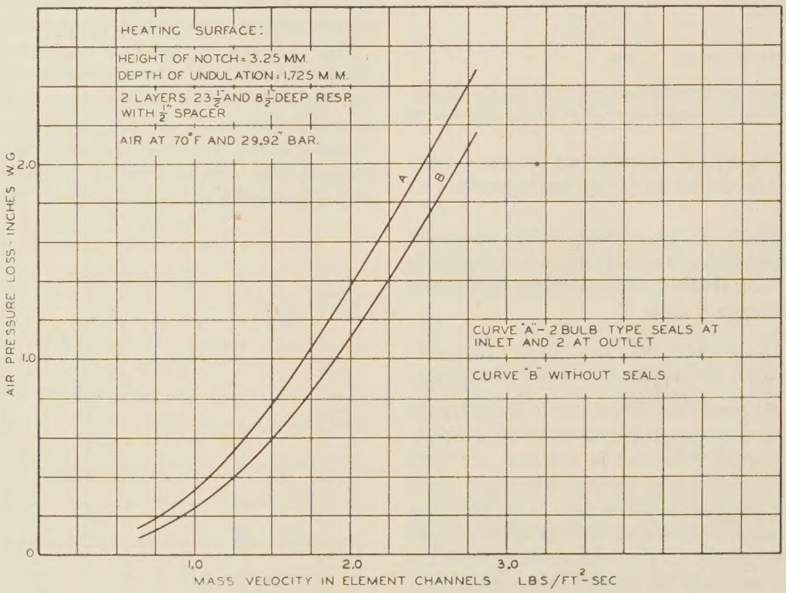

Fig. 22 Pressure Drop for Notched Undulated Heating Surface

of seals to the cross-sectional area of the duct in this arrangement corresponds closely to the highest ratio actually used in the design of a Ljungstrom air preheater. The curves cover the pressure drop through the heating surface and seals only, but do not include additional losses such as losses in inlet and outlet connections, acceleration losses, etc.

Tests were also run to determine the effect of different spaces between two or more layers of elements. It was found that the space between the two layers up to 3 in. had no measurable effect on the fluid resistance.

The curves shown in Fig. 23 give the effect of different hydraulic diameters on the fluid resistances based on these tests.

Results of pressure-loss tests on actual preheater installations show a higher frictional resistance through the units than anticipated from laboratory tests. This discrepancy is due to a less favorable cross section of flow in the rotor than in the experimental test section. Impact losses, caused by protruding edges at joints and abrupt changes in area and shape in duct design at the preheaters, also tend to increase pressure drop.

General Comments

In the foregoing the theories and various factors affecting the heat transfer and fluid resistance in a rotating regenerative type of air preheater have been given. The heat-transfer rate obtainable as well as the fluid resistances vary greatly with the type and design of heating surface employed. The subject is covered in a report presented at the World Power Conference in Tokio in 1929, by Lysholm and Edenholm (2). However, in this connection,

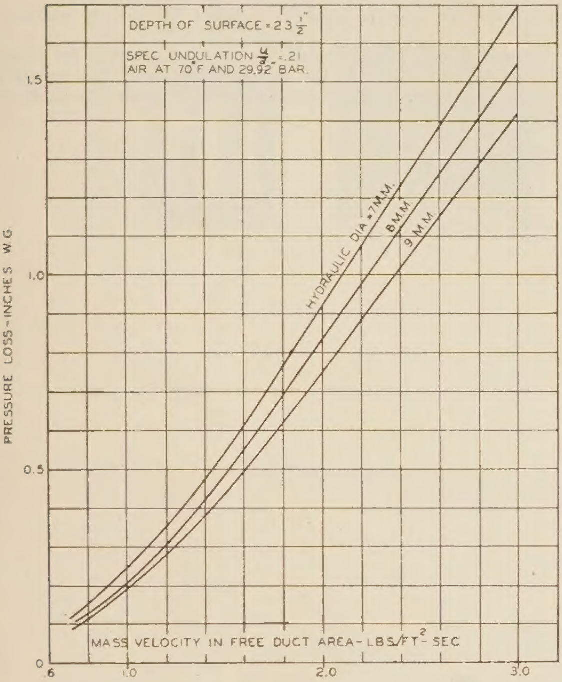

Fig. 23 Pressure Loss in Elements as a Function of Hydraulic Diameter

the authors wish to point out some of the factors affecting the actual performance of the regenerative type of air preheaters in the field.

When this type of air preheater was first applied to steam-generating units, the mass velocities employed were relatively low, as were the velocities in the duct work leading to and from the air preheater. Under these conditions, it was found necessary to apply certain corrections to the anticipated performance indicated by the laboratory tests on the rate of heat transfer as well as on the fluid resistance.

The rates of flow as used today are approximately twice those used some years ago. It has been found necessary to change further the correction factors for laboratory tests in order to make them useful in the design of air preheaters. In a laboratory model, such as that used by the investigators in Stockholm, it was impossible to duplicate or evaluate all of the conditions encountered in the field, several of which affect the performance considerably.

Actual tests of existing installations have proved that discrepancies exist in the flow of gas and air to and from the air preheater. The duct design around the air preheater is therefore of greater importance, and we are glad to report that this is a factor which, during the last few years, has been given more thorough attention.

Acknowledgment

The authors wish to express their appreciation to the Ljungstrom Steam Turbine Company in Stockholm for placing at their disposal some of the information quoted. Our thanks are also extended to the authors of other papers on the subject of preheaters, referred to in this presentation.

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Discussion

A. C. Pasini.[5] This paper is valuable in that it contributes additional technical information on the subject of heat transfer as related to preheaters; information which is not too abundant.

Unfortunately, the authors pass over the subject of air leakage rather hastily. This is a subject that is little understood. It would be beneficial if this matter could be more fully discussed in their closure.

There is considerable difference of opinion regarding the calculation of the dew point, particularly for preheater installations. Perhaps the authors would care to discuss this point, as well.

Authors' Closure

Due to limitation of the original paper, the authors passed over the subject of air leakage rather hastily, as mentioned by Mr. Pasini.

All types of air preheaters used in power-plant practice, whether of the regenerative or of the recuperative type, have some infiltration of air toward the gas side of the units. In the Ljungstrom continuous regenerative type of air preheater, the air leakage consists of entrained leakage carried by the rotor into the gas side for each revolution and the direct leakage, which is air passing over the radial and circumferential rotor seals toward the gas side of the unit.

The amount of air leakage in a Ljungstrom air preheater is quite low and varies, based on actual tests, from $3^1/_2$ to 8 points, expressed in percentage of weight of gas entering the air preheater, the entrained leakage being a very small fraction of the total.

The amount of air leakage toward the gas side of an air preheater must be considered not only in establishing its actual efficiency, as mentioned in the paper, but also in the selection of the forced- and induced-draft fans.

As to the effect of this leakage on the over-all plant efficiency, it is usually not considered that increasing the thermal efficiency of the air preheater slightly would give sufficient additional recovery to pay for the additional power consumption of the forced- and induced-draft fans required because of air leakage.

In a recent installation of Ljungstrom continuous regenerative-type air preheaters, the additional reduction in gas-outlet temperature was estimated to be 3.5 F to offset the extra power consumption of the forced-and induced-draft fans required to handle the additional air and gas quantities caused by an estimated 11.4 per cent air leakage, which additional recovery was obtained by a slight increase in the depth of the heating surface used.

In this connection, it may be proper to mention that actual field tests of Ljungstrom air preheaters show an air leakage approximately one half of the estimated.

With reference to Mr. Pasini's statement that there is a considerable difference of opinion regarding the calculation of the dew

[5] Technical Engineer, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

point, this unfortunately is true in that so many factors affect the dew point, each of which cannot be accurately predicted. Hence, the methods used by most air-preheater manufacturers in arriving at dew points are based upon practical observations and are, therefore, comparative. For a complete discussion of this subject, the authors refer to a paper by Joseph Waitkus.[6]

[6] "Operation and Maintenance of Air Preheaters," by Joseph Waitkus, Jr., presented at the Spring Meeting of the American Society of Mechanical Engineers, Houston, Texas, March 23–25, 1942.